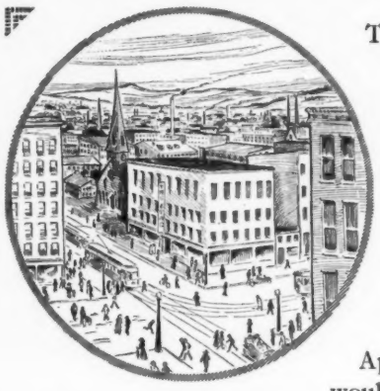
# POWER

Vol. 41 NEW YORK, APRIL 20, 1915 No. 16


## The Seventy-Fourth St. Turbine



The condensing water daily required by the turbine is as great as the daily water consumption of any one of the following cities: Hoboken, N.J.; Manchester, N.H.; Saginaw, Mich.; Binghampton, N. Y.; Charleston, S. C.; Galveston, Tex.; Norfolk, Va.; Stamford, Conn.; Chattanooga, Tenn.; Woonsocket, R. I.; Superior, Wis.

Approximately 100,000,000 gal. of water would be required for the condenser for a 24-hr. run at the rated load of 30,000 kw. The cross-sectional area of the intake tunnel is more than 138 sq. ft.

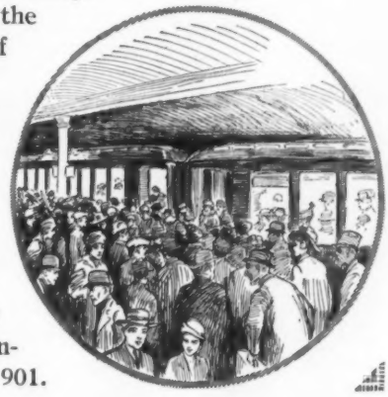
AT 1.30 lb. coal per kw.-hr. and 24 hr. a day service, the turbine will use 504 tons of coal. It would require a train of 14 cars, each carrying about 35 tons, to haul a day's supply. A good miner would be 51 days getting out enough coal to last the turbine unit one day.

*One of the Westinghouse "Cross-Compound" Turbines for the 74th Street Station of the Interborough Rapid Transit Co., New York City.*

IF the condenser tubes were slit and flattened out there would be enough metal to completely cover over an acre of ground. Placed end to end the tubes would extend 36 miles. The tubes are each 1 in. diameter, No. 18 gage and are all of admiralty composition.

The turbine will furnish power for electric railway traffic. It can haul at one time a line of people 36 miles long, allowing a space of only two feet between each person. Used solely for illumination with arc lamps it could light an area of 40 sq. miles with lamps arranged as in lighting city streets. For the same output the higher economy of each of these turbines would effect a saving of approximately 240 tons (2000 lb. each) of coal every 24 hr. over the consumption by the engines which they displace.

Based on 24-hr. operation and with coal at $3.00 per ton this would mean a saving of over $700 a day for each turbine. The old Manhattan-type engines were installed in 1901.



# Seventy-Fourth Street Station and Its New Cross-Compound Turbines

By Charles H. Bromley

*SYNOPSIS—What the installation of 30,000-kw. cross-compound turbines has meant for the Seventy-Fourth Street station of the Interborough Rapid Transit Co., New York City. Many excellent views of the turbines, and of the station during their installation, are shown. A table of ratios and important data concerning the station forms a valuable part of the article. See also the Foreword in this issue.*

"The 5000-kw., direct-connected units now being built by the E. P. Allis Co., of Milwaukee, for the main power station of the Manhattan Railway Co., at Seventy-Fourth and Seventy-Fifth streets on the East River, New York City, are the most powerful steam-operated machines of which we have any knowledge outside of the engines of the great ocean steamers."

So reads the opening paragraph of the leading article in Power for June, 1901, which describes the units installed in the Seventy-Fourth Street station of the present Interborough Rapid Transit Co. These units were the last word in large stationary steam engines. Yet three have been, and a fourth is being, broken up for junk despite the fact that they were in as perfect physical condition and that their economy was as good, as a month after installation.

## A Notable Example of Obsolescence

Amid the din of rock drills, hammer blows, rumbling cranes, the hum of turbines and the clack of releasing valve gears, this station is rapidly developing from the old to the new. Yet it is little more than a decade since these units were installed. Such is the rapidity of power-plant progress, such the ruthlessness of obsolescence.

As is generally known, these old units have two cross-compound engines each—the low-pressure vertical, the high-pressure horizontal—connected to a common crankshaft. The rated capacity of each is 8000 hp., but at one-third cutoff, 150 lb. initial pressure, 26 in. vacuum and 75 r.p.m., each can develop a maximum of 12,000 hp. The guaranteed steam consumption (dry saturated), under the above condition but for normal rating, was 13 lb. per i.hp.-hr. Today the consumption is nearly the same, being 17.3 lb. per kw.-hr.

## Why Low-Pressure Turbines Were Not Installed

The economy and the excellent physical condition of the units have caused many to wonder why low-pressure turbines were not connected to them, as practiced with such satisfactory results at the company's Fifty-Ninth Street station. Briefly, the chief reasons are these: First, the economy of the turbine as a prime mover at the time of the Fifty-Ninth Street installation was not nearly as good as at present. Secondly, the engines at Fifty-Ninth Street, in addition to being in excellent physical condition, were designed for a higher pressure than those at Seventy-Fourth Street, and, quite important, they have poppet valves in the high-pressure cylinders, adapting them to high-pressure superheated steam, while this advantage is not possessed by the Corliss-valve units at Seventy-Fourth Street. Thirdly, the complete expansion turbines (turbine and generator combined) now going in at Seventy-Fourth Street were bought at a comparatively low figure—about one-third the price per kilowatt paid for the engine units. Fourthly, it is necessary to economize on space at Seventy-Fourth Street, and complete expansion turbines accomplish this far better than combination units. For a more exhaustive analysis of the reasons for the selection of low-pressure turbines for the Fifty-Ninth Street station and of complete expansion turbines for Seventy-Fourth Street, see the article by the writer in Power, Mar. 24, 1914, page 398.

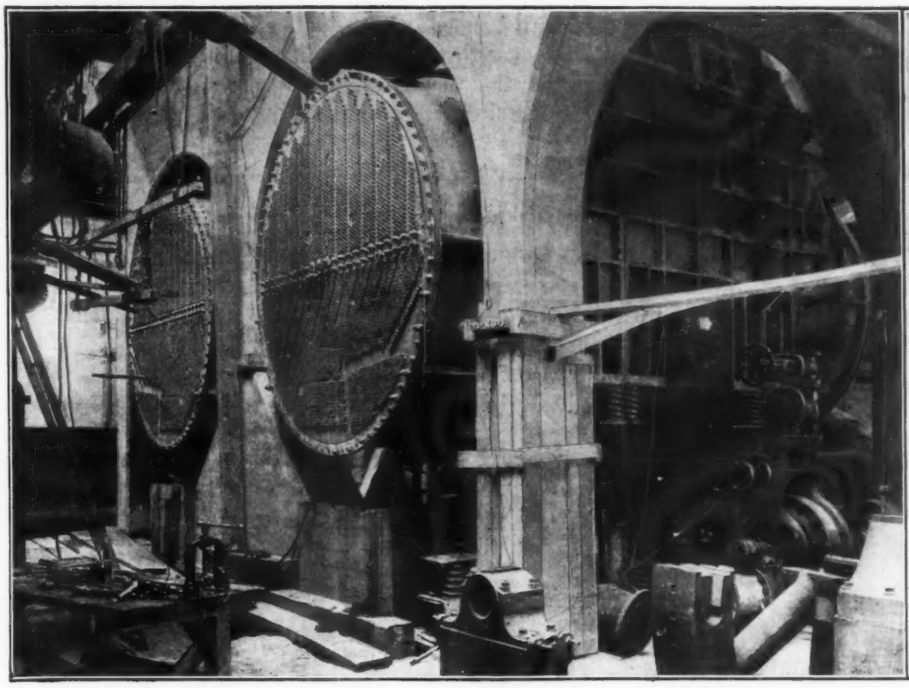


FIG. 1. NOTE THE STEAM PASSAGES IN THE TUBE BANK. CONDENSERS SUPPORTED ON SPRINGS MOUNTED ON SCREW JACKS

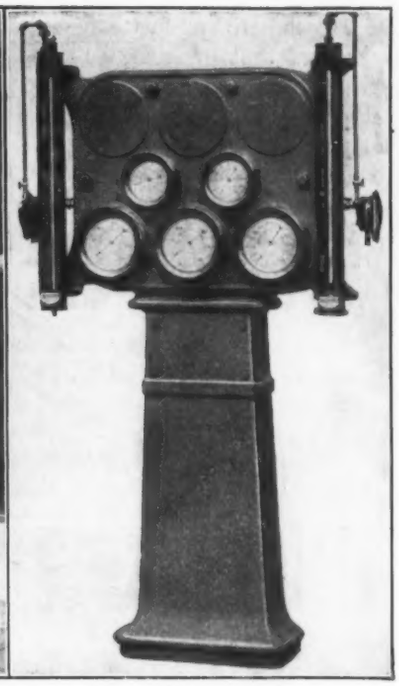


FIG. 2. THE GAGE BOARD FOR ONE OF THE 30,000-KW. TURBINES

As an indication of the advance in the economy of prime movers in recent years, the water rate of the new turbines for Seventy-Fourth Street is 30 per cent. better than for the engines they displace. For 30,000 kw. output per day of 24 hr., the turbine would save about $700 in coal alone.

### WHY CROSS-COMPOUND TURBINES WERE SELECTED

The chief feature of these 30,000-kw. units is that they consist of two turbines, a high- and a low-pressure, erected side by side. Each half drives a generator, the high-pressure running 1500 r.p.m. and the low-pressure 750; the generators are tied together electrically. The turbine is of the reaction type throughout, no impulse wheel being used. The high-pressure element is single-flow, while the low-pressure is a double-flow machine. By dividing the unit into two unconnected parts the heat drop in each casing is also divided, which eliminates the distortions and the consequent severe stresses feared in large turbines having single casings. Most important, however, is the fact that by using two speeds the relations of steam velocities to blade speeds may be correctly met in both the high- and the low-pressure ends. The double-flow principle as applied to the low-pressure end also obviates the necessity of dummy pistons to balance the end thrust. The advantages of the cross-compound principle from the designers' standpoint have been so well brought out in an article written especially for POWER (Sept. 14, 1914, page 374) by Francis Hodgkinson,* designer of these units, that the reader is referred to it for further particulars.

### BOILER PRESSURE INCREASED AFTER THIRTEEN YEARS' SERVICE

Until shortly before the installation of the new turbines, the Seventy-Fourth Street station furnished dry saturated steam to the engines at 160 lb. pressure. As the turbines are to have an exhaust pressure maintained at 97 per cent. vacuum (29.1 in., or 0.442 lb. absolute), it was desired to obtain the advantages of a higher initial pressure than 160 lb. So the advisability of increasing it to 215 lb. was considered. The boilers had been designed

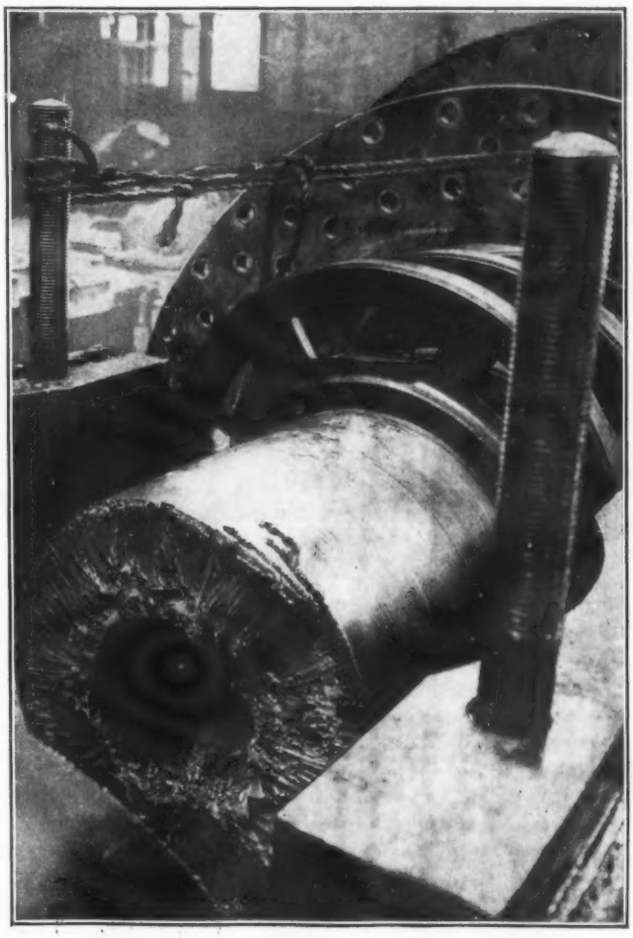


FIG. 3. THIS 32-IN. SHAFT, 16-IN. HOLE, CUT THROUGH IN THREE HOURS WITH OXYACETYLENE TORCH

*Engineer, turbine department, Westinghouse Machine Co.

for 212 lb., with a factor of safety of 5, though never worked at that pressure, and had given thirteen years' service.

Pieces were cut from the steam drums and subjected to physical tests, which showed that the tensile strength is greater now than called for in the original specifications, 55,000 lb. having been specified, while the tests showed the present strength to be from 64,000 to 69,000 lb.

It is interesting to know that micrometer measurements of the plates show them to be from 0.01 to 0.02 in. thicker than the original specified dimensions.

After an investigation of all factors affecting safety, the owners and the Police Department decided that 215 lb. pressure was permissible, and a well-known insurance company assumed the risk at normal premium rates. In the calculations used to determine if 215 lb. would be an allowable working pressure, a factor of safety of 5 was used. All cast-iron mud drums were replaced with wrought steel, and steel fittings were put in to replace the cast-iron ones removed.

FIG. 4. ONE OF THE CIRCULATING WATER PUMPS. CAPACITY 37,500 GAL. PER MIN.

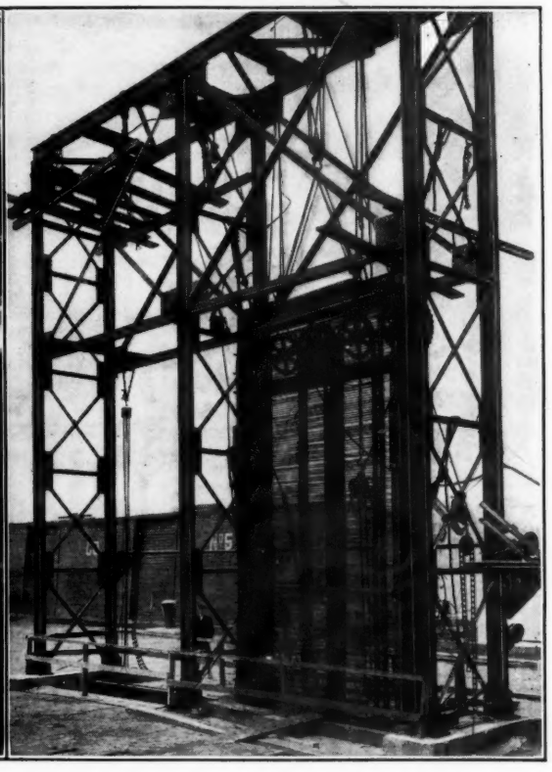


FIG. 5. REVOLVING SCREENS AT THE CONDENSER WATER INTAKE TUNNEL

### CHANGES MADE IN THE BOILER ROOM

The need for higher rating of the boilers at greater economy was the deciding factor in the removal of the overfeed stokers and the installation of those of the underfeed type. With the former and the old engines 1150 kw. per stoker was the permissible maximum attainable, while with the latter and the new turbine 3750 kw., or 225 per cent. more, may be satisfactorily carried. The engineers for the purchaser state that the furnace efficiency for the underfeed stoker is 10 per cent. greater than for the overfeed in average running. The boilers will operate at 300 per cent. rating during the peaks, which come twice a day, each lasting about two hours.

As the boilers run at higher rating than formerly and, consequently, need more flue cross-sectional area, and partly because the feed temperature is high with all-steam auxiliaries, the economizers were removed. This enlarged the effective flue area enough to increase the natural draft from 0.7 to 1.25 in.

### MAY ATTEMPT TO COOL FIREBRICK

It is the company's practice to set the firebrick in the furnace side walls from the grate to a point a little above the fire line, so that they may be renewed without disturbing the rest of the brick. This is true of all refractory material in contact with the fire. Heretofore, the same quality brick has been used below and above the fire line. But as the furnace and fuel-bed temperatures with the underfeed stokers will be higher than with the overfeed, it may be found expedient to use a better grade below the fire line. In the hope of using a low-grade, low-priced brick without experiencing the usual troubles, the experiment of embedding in each side wall a pipe carrying exhaust steam and air which will be discharged through small holes in the walls, is being tried out.

### RATIO KILOWATTS TO BOILER HORSEPOWER

There are 70 boilers in the Seventy-Fourth Street station, six of 600 hp. each, the remainder, 520 hp. Eight of the latter were allowed for each of the 7500 kw. maximum capacity, reciprocating engines, giving a ratio of 1.8 kw. per rated boiler horsepower, installed capacity. It is worthy of note that eight of these boilers will be allowed for each 30,000-kw. turbine unit (see footnote under table, page 531), a ratio of 7.2 kw. per rated boiler horsepower installed, during peaks—the highest yet practiced, though it should be understood, of course, that an equivalent output is sometimes reached in other large plants during peaks. Unfortunately, at this time an economy,

# Important Data, Seventy-Fourth St. Station

## of the Interborough Rapid Transit Co., New York City, 1915

### BOILERS

| | |
|---|---|
| Total number of boilers | 70 |
| Heating surface, each, square feet— | |
| Six | 6000 |
| Sixty-four | 5200 |
| Number of boilers with superheaters* | 32 |
| Superheating surface per boiler, square feet | 968 |
| Grate surface per boiler, square feet | 92 |
| Number sq.ft. heating surface per sq.ft. grate surface | 56.52 |
| Number sq.ft. superheating surface per sq.ft. grate surface | 10.52 |
| Number sq.ft. heating surface per sq.ft. superheating surface | 5.37 |
| Heating surface per connected kilowatt, sq.ft. | 1.386 |
| Kilowatts per sq.ft. superheating surface | 3.88 |
| Kilowatts per sq.ft. grate surface | 40.76 |
| Kilowatts per boiler horsepower, installed capacity | 7.21 |
| Type of boilers | Babcock & Wilcox |
| Designed for 212 lb. pressure with factor of safety of | 5 |
| Present pressure, lb. | 215 |
| Boilers now have wrought-steel mud drums. | |
| Boiler rating on peaks, 300 per cent.; between peak, about 100 to 150 per cent. | |
| Underfeed stokers, seven-retort "Taylor," | American Engineering Co. |
| Capacity of each stoker on peaks, kw. | 3750 |
| Chimneys— | |
| Number | 4 |
| Height, above lower grate, ft. | 261 |
| Diameter, inside, bottom, ft. | 18 |
| Diameter, inside, top, ft. | 17 |
| Boilers per chimney | 16 |
| Coal burned per sq.ft. grate, lb. per hr.— | |
| At normal rating | 19 |
| At maximum rating | 60 |

*Only 32 boilers would be required and would be actually used to supply steam to three 30,000-kw. turbines, and of these 32 boilers it is anticipated that not less than four would be out of service continuously for repairs, overhauling, etc. The ratios are based on eight boilers per 30,000-kw. turbine, however.

### TURBINES

| | |
|---|---|
| Three 30,000-kw. "cross-compound" in present installation | |
| Builders | The Westinghouse Machine Co. |
| Speed: High-pressure, 1500 r.p.m.; low-pressure, 750 r.p.m. | |
| Generators: Each, 15,000 kw., 3-phase, 25-cycle, 11,000 volts. | |
| The two generators of each unit are tied together electrically. | |
| Pressures: High-pressure initial, 200 lb.; initial pressure of low-pressure side, 12 lb. abs. at 16,000 kw.; 15 lb. abs. at 25,000 kw.; 19 lb. abs. at 30,000 kw. | |
| Superheat at throttle, deg. F. | 120 |
| Vacuum, 97 per cent., or in inches of mercury | 29.1 |
| Performance guarantees: Operating conditions—200 lb. gage pressure, 120 deg. F. superheat and 29 in. vacuum (referred to a 30 in. barometer). | |

| Net Kw. Load of Generator | Lb. of Steam per Kw.-Hr. | Rankine Cycle Efficiency, per Cent. |
|---|---|---|
| 15,000 | 12.07 | 70.73 |
| 16,000 | 11.94 | 71.51 |
| 18,000 | 11.77 | 72.54 |
| 20,000 | 11.54 | 73.98 |
| 22,000 | 11.40 | 74.89 |
| 24,000 | 11.30 | 75 56 |
| 25,000 | 11.27 | 75.76 |
| 26,000 | 11.32 | 75.42 |
| 28,000 | 11.47 | 74.44 |
| 30,000 | 11.63 | 73.41 |

| | |
|---|---|
| Most economical load, per cent. of max. 24-hr. load | 83.3 |
| Steam consumption of auxiliaries, per cent. main unit consumption— | |
| At most economical load | 7.5 |
| At full load | 6.08 |
| Power consumption of auxiliaries, per cent. main unit power, at full load | 1.4 |
| Blading, reaction; bronze throughout. | |
| Peripheral speed last rows low-pressure, ft. per sec. | 400 |
| Total weight, lb. | 1,500,000 |
| Weight per kilowatt, lb. | 50 |
| Heaviest piece to be lifted by crane, tons | 78 |
| Floor space, outside measurement, sq.ft. approx. | 1600 |
| Kilowatts per sq.ft. floor space | 18.7 |

### CONDENSERS

| | |
|---|---|
| Builder | Henry R. Worthington Co. |
| Total tube surface, sq.ft. | 50,000 |
| Tubes, admiralty, 1 in. diam., 20 ft. 3⅝ in. long, 18 gage. | |
| Chief guarantee, 350,000 lb. steam condensed, water at 60 deg. F., 65,000 gal. water per min., maintaining 97 per cent. vacuum, 29.1 in. mercury. | |
| Tube area per kw., sq.ft. | 1.67 |
| Circulating pumps: Two per condenser, centrifugal; capacity each, gal. per min. | 37,500 |
| Type | Twin-shell, counter-current |
| Circulating-water pumping capacity per kw., gal. per hr. | 150 |
| Circulating-water pumping capacity per lb. steam condensed at consumption of 350,000 lb. per hr., pounds | 107 |
| On guarantee, 65,000 gal. per min., lb. | 93 |
| Diameter discharge pipe, in. | 60 |
| Cross-sectional area intake tunnel, sq.ft., approx. | 138 |
| Revolving, self-cleaning screens in intake. | |
| Maximum speed of tide in river, miles per hr., approx. | 7 |
| Area of each exhaust in condenser, sq.ft. | 142 |
| Steam velocity through each exhaust opening, ft. per sec. | 227 |
| Reciprocating dry vacuum pump, size, in. | 14x39x30 |
| Maker | Laidlaw-Dunn-Gordon Co. |

### MISCELLANEOUS

| | |
|---|---|
| Turbine foundations of structural steel. | |
| Kilowatts per cu.yd. concrete in foundation | 109 |
| The foundations for the 7500-kw., max. 24-hr. rating engines each had 1500 cu.yd. of concrete. | |
| Anticipated station load, kw. | 100,000 |
| Total upward pressure of atmosphere on condenser when carrying 29 in. vac., lb. | 290,566 |

or performance-at-different-load, curve of this turbine is not available for publication. It may be said, however, that it is liberally designed, for tests show that the consumption at full load is but little higher than that at the most economical load. Also, the turbines—three are to be installed for the present—will each easily carry 32,000 to 33,000 kw. with but a slight increase in steam consumption. The two now in service each carry this load nearly every morning.

### Rubber Expansion Joints in Circulating Water Pipes

There are two condensers, one connected to each low-pressure exhaust outlet, and each is in three sections. The total tube surface is 50,000 sq.ft., in 4780 admiralty tubes, each 20 ft. 3⅜ in. long, 1 in. diameter and of 18 gage. The chief guarantee is 350,000 lb. steam condensed per hour with water at 60 deg. F., maintaining a vacuum of 97 per cent. (29.1 in., or 0.442 lb. absolute). The steam opening into each condenser is 142 sq.ft. in area. The openings in the bank of tubes to assure steam getting down around the bottom tubes are plainly shown in Fig. 1. Fig. 2 shows the gage board used.

The circulating-water pipes which supply both condensers are 60 in. diameter. A novel feature of construction is that there are no expansion joints between the turbine and condenser, each condenser being rigidly bolted to one of the exhaust flanges of the double-flow low-pressure turbine, there being provided a 36-in. connection with a copper expansion joint between the two condensers to maintain equilibrium of pressure. Hence, to provide for the expansion and contraction of the turbine under different operating conditions, it is necessary that the condensers be able to translate themselves with reference to each other. This necessitates quite flexible expansion joints between the circulating pipes and the water chambers. Copper expansion joints were first installed, but it was found that these held the condensers too rigidly and that they would not move with the expansion and contraction of the turbine. This difficulty was overcome by substituting rubber expansion joints. Inasmuch as the piping was in place, the joint must be so designed as to make use of the flanges and, to avoid special rubber work, must admit of using plain sheet rubber instead of a molded piece. The rubber is of a good grade and is five-ply, ½ in. thick and made up similar to belting.

Reference to Fig. 1 will show that the condensers are supported on heavy springs resting on screw jacks.

The air pump is of the reciprocating kind, being 14x39x30 in.

Fig. 4 shows one of the circulating pumps and Fig. 5 the revolving type of screen used at the intake.

### Structural-Steel Supports

The economy in the use of concrete by using structural-steel supports for turbines over the old solid engine foundations is well demonstrated at Seventy-Fourth Street. The foundations for each of the 7500-kw. engine units required 1500 cu.yd., while for each 30,000-kw. turbine supported on structural steel, but 275 cu.yd. is needed and this chiefly to stiffen the supports against vibration. This is the heaviest turbine yet to be supported on structural steelwork. Although the tandem-compound, 30,000-kw. unit in Waterside No. 2 is so supported, it weighs less. The condenser support of the Waterside structural work has a system of spring beams to avoid the necessity of an expansion joint between the turbine exhaust opening and the condenser steam inlet.

In the removal of the large engines it was found expedient to cut the main shafts. These are each 37 in. diameter with a 16-in. hole, and by employing the oxy-acetylene torch it took but three hours to make a cut. See Fig. 3.

### The Construction of the Turbine

The high-pressure side contains 38 rows of blades and differs but little from any other single-cylinder reaction turbine. The first 8 rows are mounted on a ring bolted to the casing. Following this are 19 rows mounted on a second and longer ring, or barrel. The remaining 11 rows are mounted directly on the casing. The rotor is 20 ft. 4⅝ in. long. The views on the insert give a good idea of the construction and size of one of the units.

The low-pressure turbine is double-flow. The casing is a simple shell affair except that it has some interesting reinforcing members. There is no blading mounted directly on the casing, but instead, it is all put on rings or barrels bolted to the casing. One of the intermediate-pressure rings for the low-pressure machine having the blading mounted is also shown in the insert.

The low-pressure rotor is made up chiefly of a hollow drum secured to two spindles, one at each end. The intermediate-pressure blades are mounted directly on the drum, and the low-pressure blades are put on rings which slip up over the spindles and are bolted fast. The spindle castings with the risers weigh 72,500 lb. each and are of steel. The rough ends without the risers weigh 38,500 lb. each.

## A Cement for Leather

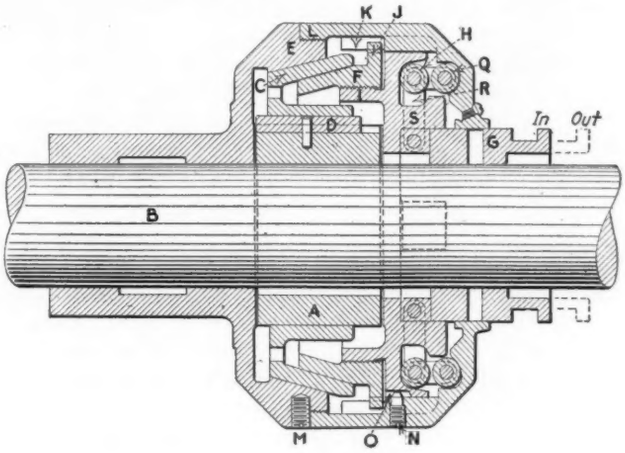
To prepare a cement suitable for leather belts or for fastening paper covering to pulleys, get the best cabinet maker's glue, in a quantity suitable to your requirements. A large quantity can be prepared if desired, for it will keep for some time after being mixed, if not permitted to dry out. Break the glue into pieces and put in a dish, with water just sufficient to cover it, and let stand twelve to fourteen hours, or until all of the water has been soaked up. Then melt the glue in a water or steam bath and add strong vinegar to thin it. It should then be evaporated until it will appear quite stringy from the stick or spoon used to stir it while hot.

A leather belt should be roughed or furred with sandpaper or a coarse file, which will make the joint stronger than if the leather were left smooth. If the leather is warmed before applying the glue, a better joint can be made. The laps should be scraped down to a thin edge and their length should be equal to the width of the belt. In making the joint, lay the belt on a board so that the parts come even, then fasten with a couple of nails through the leather a foot or more from the joint on each side, so that when one piece is raised from the other to apply the glue, it will fall back into the proper position. Apply the glue warm and pound the joint all over with a hammer and a block laid on the leather. A few tacks driven through the joint will assist in holding it together properly until the cement has set.

The same preparation can be used for fastening paper, cloth or split leather to a pulley to increase the driving power of the belt. If of iron, the pulley should be well cleaned by scraping and then washed with strong vinegar or a weak solution of sulphuric acid in water. Wipe dry and apply the paper, which has previously been covered with the hot glue. Two or three thicknesses of heavy straw paper will be found sufficient, and each layer should be firmly glued on. As soon as the first layer is applied and before the glue has had a chance to cool, roll or hammer the paper to bring it in contact with the pulley. Each layer should be treated in the same way. A belt should remain undisturbed for about five hours after the joint is made, before an attempt is made to use it. Three or four hours will be sufficient for the paper covering on the pulley.

## Ideal Multi-Cone Clutch

This clutch, which is manufactured by the Akron Gear & Engineering Co., Akron, Ohio, might be termed a multi-disk cone clutch. In its design it retains the simplicity of the two-cone clutch, allows smooth engagement due to momentary slipping, and will release instantly. To avoid a too sudden engagement of the cones their face angles are greater than those of the ordinary two-cone type, and the clutching force and pulling power so lost are more than compensated for by the addition of a third cone, which practically doubles the pulling power. The cones run in an oil bath which leaves a film between them and permits a slippage before being broken down by the pressure of coming into engagement. Owing to the face angles and the small unit pressure on them, as well as the oil bath, immediate disengagement occurs when the clutch is thrown out.




IDEAL MULTI-CONE CLUTCH

Means are provided to prevent the oil from escaping to the outer surfaces where it would be thrown off by centrifugal force. The horizontal pressure exerted by the throw in the mechanism is distributed equally around the circumference and does not distort the cones from a true circle.

When the clutch is out the throwing mechanism is still and centrifugal force cannot throw it in. When it is in centrifugal force cannot throw it out, but will tend rather to keep it in.

Referring to the illustration, the driving ring *A* is keyed to the shaft at *B*. The middle or driving cone *C* is driven by the ring *A* through two feathers *D;* both friction surfaces of this cone contain oil grooves. The driven cones *E* and *F* are brought into contact with *C* when the shifter sleeve *G* is pushed in, thus throwing the rollers *H* outward and forward against the adjustment ring *I*, which carries the cone *F* forward into contact with *C*, and the latter into contact with *E*. As the cones come into contact singly, too sudden clutching is prevented. The cone *F* is caused to revolve with *E* by means of the lugs *J* projecting outward on *F*, which lie between the lugs *K* on the casing *L*. The inner faces of the lugs *K* are turned true and hold the ring *I* central. The casing screws on the cone *E* and is locked by the screw *M*. The inner end of the locking screw *N* projects into one end of the numerous slots *O* in the outside periphery of the ring *I* and causes it to revolve within the casing *L*. The adjustment of the clutch is made by inserting the screw into one of the slots. The rollers *H* and *Q* and their pivot pins are large in diameter and in bearing area. The links *R* straddle the lugs *S*, the ends of which are raised slightly.

In throwing the clutch out, the rollers *H* strike the lugs *S* and pull the cone *F* forward, which gives the maximum clearance for the oil films between the cones. The throwing mechanism is powerful, the multiplication between the horizontal force on the shifter sleeve *G* and the pressure on the cone faces being approximately 100 to 1 on all sizes.

## Large Buffalo Transformer Blower

An unusually large fan for cooling air-blast transformers has recently been installed at the Blue Island Power Station, Public Service Co. of Northern Illinois, by the Buffalo Forge Co., Buffalo, N. Y. The installation was made under the direction of Sargent & Lundy, engineers. It consists of a direct-connected blower handling continuously 40,000 cu.ft. of air per min. at 70 deg. F. and 29.92 in. bar., with a static increase in pressure of 2.6 in. water gage. The blower is directly connected to a 30-hp., 25-cycle, three-phase, 470 r.p.m. motor. Aside from the size of the unit, the interesting feature is the operating speed.

Most transformer cooling units are 20,000 cu.ft. per min. capacity or below, and although direct connection is desirable, it has heretofore involved prohibitive expense for the slow-speed motors necessary on larger units. The blower in this case is a turbo conoidal high-speed type, such as has been used in connection with motor and steam-turbine-driven forced-draft units for underfeed-stoker work. Although the air pressures required for cooling air-blast transformers are considerably less than for stoker work, the speed of this fan is high enough to permit the use of a motor at a price which is not excessive. The fan is of the multiblade type with compact housing. The photograph shows the relative sizes of the fan and the motor. The fan has a static efficiency of 60 per cent. and requires 27 b.hp.

MOTOR-DRIVEN FAN OF 40,000 CU.FT. OF AIR PER MIN.

# Theoretical Efficiency of Heat Engines

By R. C. H. Heck

*SYNOPSIS—Distinction between "actual" and "ideal" efficiencies, explanation of the Carnot efficiency and when it should be applied, also the error involved in using the "air standard" in connection with the Carnot efficiency in internal-combustion engines.*

The term "heat engine" covers all forms of apparatus for converting heat into work, and the only practical way of making this conversion is by means of an expansive medium which may be either a liquid (alternately vaporized and condensed) or a dry gas mixture. In judging the performance of a heat engine there are three efficiencies to be considered—actual, ideal and relative.

Actual efficiency is the ratio of the heat converted into work (or work output measured in heat units) to the heat supplied, these quantities being determined by test or experiment. No heat engine converts into work more than a minor portion of the heat energy which it receives. This is due largely to the inherent nature of the heat-engine process and in lesser degree to imperfections of actual material and operations. For any set of limiting conditions it is possible to calculate, from thermodynamic theory, the efficiency of an ideal heat engine, free from all secondary imperfections.

This ideal efficiency is the proper basis for judgment of actual performance. Thus, a conversion ratio of 0.24 for the best steam turbine may seem a poor showing, but when theory determines that an ideal apparatus under the same conditions could convert only 0.34 of the heat received, it appears that the real engine is doing about 70 per cent. as well as the ideally perfect one. In medium to very good practice this relative efficiency, the ratio of actual to ideal, ranges from 50 to 75 per cent.

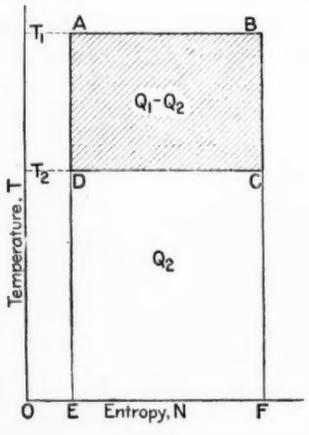



Fig. 1

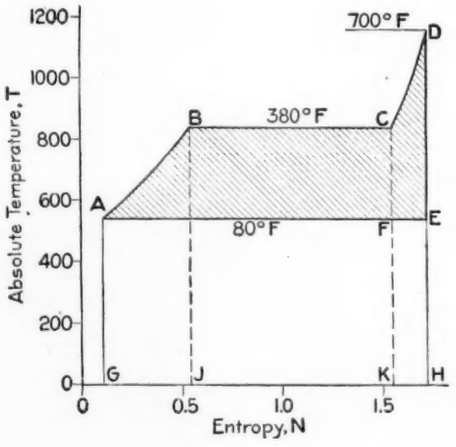



Fig. 2

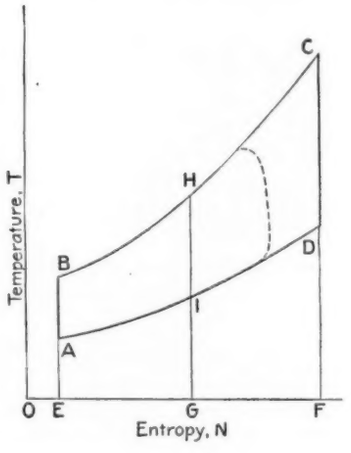



Fig. 3

## The Carnot Engine

Any heat engine operates by receiving heat at high temperature, converting a part of it into work, and rejecting the remainder as heat at low temperature. The simplest possible scheme, thermally, is that of the ideal Carnot engine, in which the temperatures of heat reception and heat rejection are constant throughout the respective operations, and in which there are no losses by radiation, cylinder-wall action or machine friction.

The temperature-entropy diagram in Fig. 1 represents the working of the Carnot cycle. The expansive medium is confined in a suitable cylinder with a piston, and at $A$ it has the upper temperature and a high pressure. Receiving heat at the constant absolute temperature $T_1$, it changes to state $B$ by isothermal expansion. The quantity of heat $Q_1$ received is represented by the area $ABFEA$; which has the width $EF = Q_1 \div T_1$. This horizontal distance $EF$ is known as the "difference in entropy" and will be represented by $N$. Applied to steam, $EA$ represents the absolute temperature corresponding to the given pressure; the condition at $A$ would be hot water at this temperature, and the area $ABFEA$ would represent the latent heat necessary to change the water into dry saturated steam, which condition is represented at $B$; finally, $AB$ $(= EF = N)$ is the distance necessary to make the area $ABEFA$ proportional to the latent heat at temperature $T_1$.

At $B$ in Fig. 1 the supply of heat is shut off and expansion continues in the perfectly nonconducting cylinder until the temperature is lowered to $T_2$ at $C$. The drop in temperature is due to the expenditure of heat energy in the work of expansion; but since no energy is transferred in thermal form, there is no change of entropy. A no-heat-transfer operation is called adiabatic.

From $C$ to $D$ the medium is compressed isothermally (at constant temperature) at $T_2$, surrendering the heat $Q_2 = NT_2 =$ area $DCFED$, and adiabatic compression from $D$ up $A$ completes the cycle. With a heat input $Q_1$ and heat, $Q_1 - Q_2$, converted, the efficiency is

$$E = \frac{Q_1 - Q_2}{Q_1} \qquad (1)$$

which is a general expression applicable to all heat en-

gines. Here $Q_1 = NT_1$ and $Q_2 = NT_2$, so that the efficiency of this cycle may also be expressed by

$$E = \frac{T_1 - T_2}{T_1} = \frac{t_1 - t_2}{t_1 + 460} \qquad (2)$$

in which $t_1$ and $t_2$ are the Fahrenheit temperatures corresponding to the absolute temperatures $T_1$ and $T_2$.

Sometimes this Carnot efficiency is used erroneously as a standard when it does not fit the conditions of the actual plant. A notable example is found in the issue of June 9 last, in the abstract of a lecture by F. G. Gasche, on "Power for Steel Mills." For a steam turbine using steam superheated to 700 deg. F. and with exhaust at 80 deg., the ideal efficiency is there computed as

$$E = \frac{700 - 80}{700 + 460} = \frac{620}{1160} = 0.535$$

As a matter of fact, this result is about 60 per cent. in excess of the correct value.

### Ideal Steam Cycle

Fig. 2 shows the ideal cycle for the steam engine or turbine, within the limits just named and with the additional datum that vaporization shall take place at 380 deg., or that the boiler pressure shall be about 195 lb. absolute. The cycle begins at $A$ with 1 lb. of feed water at exhaust temperature pumped into the boiler. Curve $AB$ represents the heating of the water up to the boiling point of 380 deg. F., or 840 deg. absolute; it receives heat and acquires entropy as the temperature rises. The horizontal line $BC$ represents the isothermal, or constant-temperature, operation of vaporization. At $C$ the steam is dry saturated, and its superheating from 380 to 700 deg. is represented by curve $CD$. At $D$ the total heat of the steam is 1370 B.t.u., and the heat imparted, beginning with water at 80 deg. F. (540 absolute), is 1322 B.t.u., represented by area $GABCDHG$.

The operation $ABCD$ is performed in the boiler, and we assume that the steam is carried over to the engine or turbine without loss of heat by radiation or of pressure by pipe resistance. Then adiabatic expansion, whether in the ideal non-conducting cylinder or in the formation and utilization of a perfect steam jet, lowers it to state $E$. Abstraction, of heat in the condenser, of the exhaust heat $Q_2$ (area $EAGHE$ = 879 B.t.u.), is represented by the isothermal line $EA$. The ideal efficiency is now, as against the 0.535 previously figured, only

$$E = \frac{Q_1 - Q_2}{Q_1} = \frac{1322 - 879}{1322} = \frac{443}{1322} = 0.335$$

The diagram shows clearly the error involved in using for $T_1$ in the Carnot expression, equation (2), the highest temperature reached by the medium, instead of making it the temperature of heat reception. When the latter temperature varies, as here, it might be replaced by a mean value of equivalent effect, although that is not the direct or the better way of calculating $E$. Just to see how nearly the vaporization temperature of 380 deg. would come to serving as such an effective average value, try it in equation (2), from which will be found

$$E = \frac{380 - 80}{380 + 460} = \frac{300}{840} = 0.357$$

Evidently $t_1 = 380$, or $T_1 = 840$, is a little too high for this purpose.

Because so much of its heat reception is at $T_1$ and all of its heat rejection at $T_2$, the steam cycle is fairly near the Carnot in general form. The practical reason for this is that the constant-pressure operations of vaporization and condensation are characterized also by constant temperature. But in no scheme of gas-engine working do isothermal operations find place.

### Otto Gas-Engine Cycle

Consider the Otto cycle outlined in Fig. 3. The operation begins at $A$ with a charge of gas mixture under atmospheric pressure and of a little higher than atmospheric temperature. Line $AB$ shows adiabatic compression, followed by the reception, at practically constant volume and with a rapidly rising temperature, of the heat of combustion. This heating along curve $BC$ is followed by the ideal adiabatic expansion $CD$, and exhaust is taken to be equivalent to constant-volume cooling along curve $DA$. The actual performance, in an actively conducting cylinder and with combustion more or less retarded, is somewhat of the form sketched by the dotted line.

It has been usual, for a simple calculation of theoretical efficiency, to assume that the gas mixture is practically the same as air in properties and that its specific heat is constant, not rising with temperature. Under the latter assumption the ratio of low to high temperature is the same on any ordinate in Fig. 3, whether at the extreme lines $BE$ and $CF$ or on any vertical $HG$. If then

$$\frac{T_A}{T_B} = \frac{T_D}{T_C} = \frac{T_I}{T_H}, \text{ etc.}$$

it is evident that the conversion area $ABCDA$ will be to total heat area $EBCFE$ as $T_B - T_A$ is to $T_B$, or that the Carnot ratio will apply if the limiting temperatures of either adiabatic are used in place of certain constant temperatures.

The principal purpose in writing this article has been to lay before the readers the fact, to be found only in the more recent textbooks, that the use of this simple air standard involves a large error. Computed results from a typical example are laid out in Figs. 5 and 6, where the dotted diagrams marked $a$ are for ideal air and the full-line diagrams $b$ represent the behavior of an actual gas mixture.

The medium selected in the example is a blast-furnace gas with a moderate excess of air in the combustible mixture. Its heat value is such that perfect combustion generates just 1000 B.t.u. per pound of mixture, the diagrams being drawn for that quantity. The specific heat rises with the temperature according to the constant-rate law shown by straight lines Nos. 2 and 3 in Fig. 4, of which No. 2 is for the original mixture and No. 3 for the products of combustion; line No. 1 shows the uniform value, $C_v = 0.169$, for the ideal air medium.

One main determinant is the pressure of 175 lb. absolute at the end of compression in Fig. 5 (points $B$ and $B'$). This and the initial atmospheric pressure and an assumed temperature of 80 deg. F., or 540 deg. absolute, at $A$ and $A'$ are the only conditions common to the two cases, although the compression curves $AB$ and $A'B'$ are very much alike. The greatest difference is seen in the temperature rise from $B$ to $C$ and $B'$ to $C''$, with the resulting expansion lines $CD$ and $C'D'$. Curve $c$ is a rough guess at the probable actual indicator diagram.

Of course, such a tremendous rise of temperature as that which carries $C'$ up to more than 7000 deg. in Fig. 6 and the corresponding pressure to 1100 lb. in Fig. 5 is

physically impossible. Even the more reasonable height of point $C$ runs into a region where dissociation is probably a potent influence and where our knowledge of specific heat is vague. But disregarding the fact that the conditions of diagram *a* are largely imaginary and those of *b* in some degree doubtful, the results of calculation may be summed up as follows:

In diagram *a*, Fig. 6, the four corner temperatures are: $T_A = 540$ deg., $T_B = 1104$ deg., $T_C = 7021$ deg., and $T_D = 3434$ deg. Then the efficiency is

$$E = \frac{1104 - 540}{1104} = \frac{7021 - 3434}{7021} = 0.511$$

In diagram *b* the corresponding temperatures are: $T_A = 540$ deg., $T_B = 1092$ deg., $T_C = 4818$ deg., and $T_D = 3315$ deg. With these the Carnot ratios at the limiting ordinates $BA$ and $CD$ are very different, being

$$\frac{1092 - 540}{1092} = 0.505 \quad \text{and} \quad \frac{4818 - 3315}{4818} = 0.312$$

A notable effect of higher specific heat is the relatively smaller vertical width of the effective area $ABCDA$, as compared with $A'B'C'D'A'$ (Fig. 5). Efficiency is found, however, not from temperatures but from heat quantities, as referred to in the method of equation (1). Along curve $BC$ (Fig. 6) the heat received from combustion is 1000 B.t.u., while that rejected along $DA$ is 632 B.t.u.; then the efficiency is

$$E = \frac{1000 - 632}{1000} = 0.368$$

The physical data for a calculation such as is represented by diagrams *b*, Figs. 5 and 6, are not complete in full accuracy. The methods of calculation are more directly related to Fig. 5, involving data as to pressure, volume and specific heat; and the heat converted, 368 B.t.u., is strictly equivalent to the work area $ABCDA$ in Fig. 5. When entropy is calculated, rather as a secondary quantity, a discrepancy develops; the diagram in Fig. 6 failing to close by the amount $A_1A$.

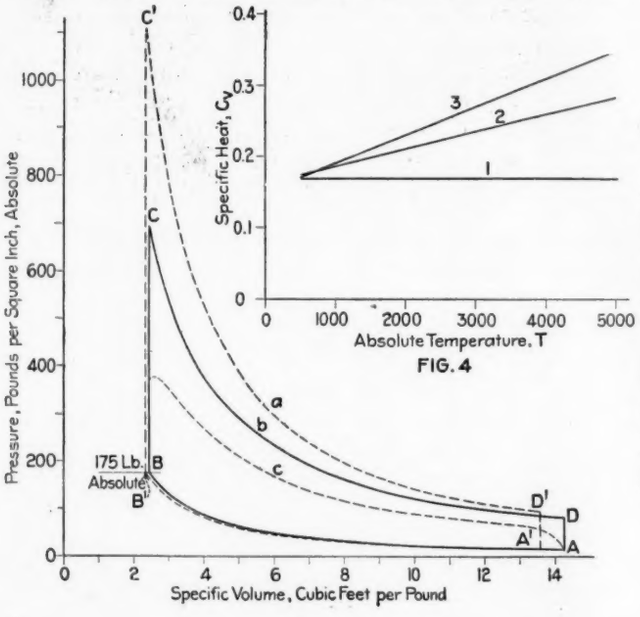



FIG. 5

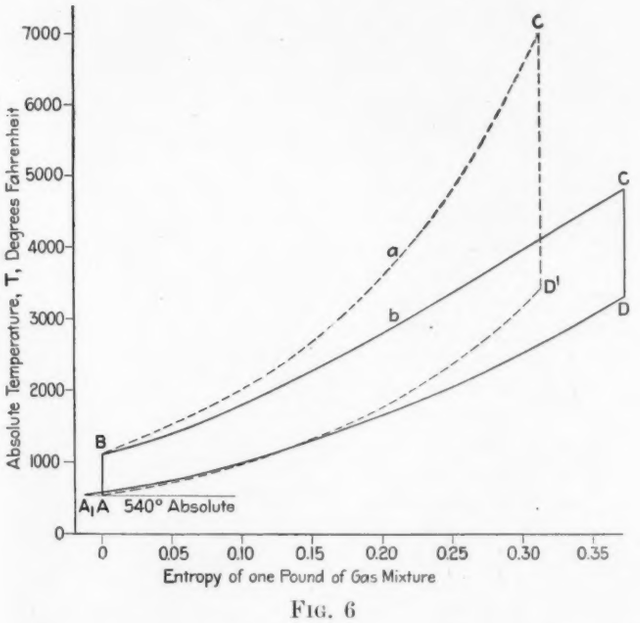



FIG. 6

## USEFULNESS OF IDEAL EFFICIENCY

Ideal efficiency, or the performance and output of the Rankine cycle represented by Fig. 2, is regularly used as a standard of comparison for steam action; but with internal-combustion engines it is little used. The simple air standard is so much in error as to be worthless, and the more correct method requires complicated and difficult calculations. For one thing, it must start in each case with the proportions of the particular gas mixture, upon which the average fundamental physical properties of the medium are dependent; and even with the simple straight-line law for variation of specific heat with temperature (itself probably no more than an approximation), the change of temperature in an adiabatic operation can be found only by a troublesome trial solution. Merely to state and outline the calculations for Figs. 5 and 6, including a number of intermediate points on each curve, would take considerable space, with no explanations or proofs; and while the direct calculation of ideal efficiency alone is much shorter, it is yet rather beyond the scope of ordinary use. Therefore, gas-power engineers will probably continue to be satisfied with actual efficiency as a measure of performance.

The really important result from the example here set forth is the relatively low value, 0.37, for true ideal efficiency, as against 0.51 under the assumption of constant specific heat. Considering the intense action of the cylinder walls in a gas engine, the relative efficiency is not likely to be over 0.6. Applying this to 0.37, we get a probable actual value of 0.22, and see that so far as the conversion of the heat supplied to it is concerned, an engine using blast-furnace gas is much in the same class with good steam engines and turbines. This does not, of course, deny the great economy of using such gas directly in the engine instead of burning it under steam boilers.

**Coal for Coke**—In the last five years the coal used in metallurgical coke manufacture has averaged around 65,577,000 tons, yielding 43,983,000 tons of coke, valued at $111,736,000. Of this total, 14,767,000 tons were used in byproduct coke ovens, yielding, besides the coke 54,491,000 cubic feet of gas 94,306,000 gallons of tar, and $9,190,000 worth of ammonia. When it is considered that every year approximately four times these enormous totals of byproducts are absolutely wasted through the use of non-byproduct ovens, the vital importance to the country of a general use of the modern, scientific byproduct ovens will be appreciated.—"Journal of the Franklin Institute."

# A Forty-Million-Revolution Compressor-Valve Test

In the endeavor to test to destruction one of the feather valves now being used by the Laidlaw-Dunn-Gordon plant of the International Steam Pump Co., for air and gas compressors, the builders fitted a small vertical compressor with one of the standard-type feather valves and with an annular valve of the so called low-lift, plate type, the latter being of standard German manu-

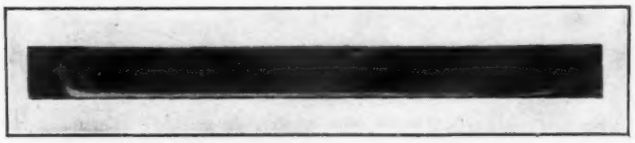


FIG. 1. SEATING SURFACE OF FEATHER VALVE. THE LIGHTER PORTION SHOWS THE SURFACE AFTER FORTY MILLION REVOLUTIONS OF THE PUMP

facture. The valves were used alternately for intake and discharge. The compressor was operated at a speed of 560 r.p.m. against a pressure of 40 lb. during each working day of ten hours, for a period of six months, aggregating in that time something over forty million revolutions. During this period three of the annular valves gave out, while the original feather valve at the end of the period had gone no further than to perfect its seat. Fig. 1 is a photograph of one of the blades used, the lighter portion indicating the polished surface of the seating area.

Forty million revolutions represents about a year's operation at 225 r.p.m. The speed of 560 r.p.m. should, on the basis of ordinarily accepted practice, result in destructive action four times as fast as a speed of 280 revolutions, which represents about the limit of commercial practice at present. The builders claim, therefore, that this run of forty million revolutions at 560 r.p.m. is the equivalent of at least four years' normal running, and the valves, judging from their appearance, had not even begun to deteriorate.

The valve on which this test was made is designated by its builders as the Laidlaw feather valve (patented),

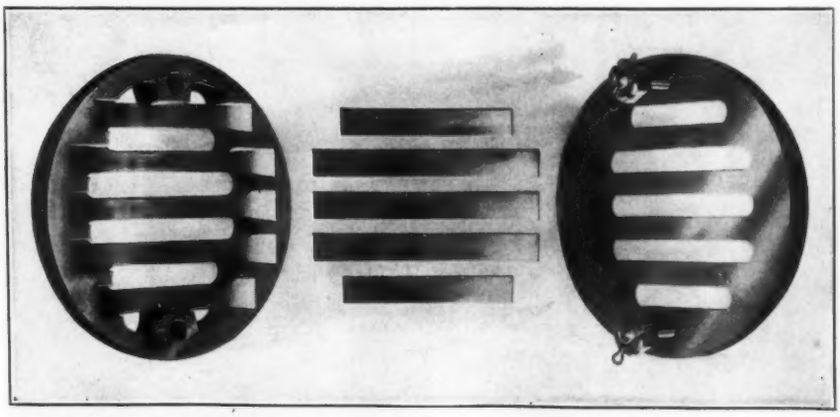


FIG. 2. THREE ELEMENTS COMPRISE THE FEATHER VALVE

Fig. 2. The complete contrivance consists of three elements only. The valves proper are strips of light flexible flat steel stock similar in appearance to ordinary clock-spring stock, but more flexible and of a much lower temper. These strips, which in the average valve are about ½ in. wide and vary in length from 4 to 12 in., seat on the ground face of a slotted casting, the slots being slightly smaller than the strips. The valves are not held rigidly at any point, but are restrained in movement by a curved guard with slots staggered to the slots in the seat, the spaces between being milled out on a curve against which the valve bows up when in an open

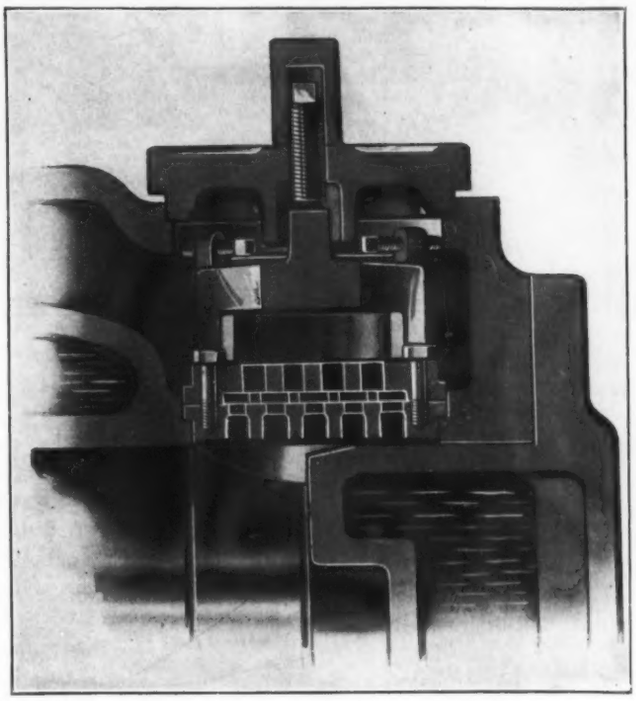


FIG. 3. DETAILS OF THE VALVE ASSEMBLED

position, to allow the passage of air. The ports are shown in Fig. 3.

The guard is lightly bolted to the seat, the strips being thus held between, free to bend up and down in the middle, their ends always remaining in contact with the seat, giving a breathing rather than a slapping action. The movement of the valve is controlled by the air flow, the valves themselves having negligible spring action, the best results being obtained with the highest flexibility. They are so light as to respond instantly to change in air flow, their flexibility not only permitting air to flow through them with a minimum of spring resistance, but also resulting, with the reversal of air travel, in almost perfect contact with the seat. The notable feature of this valve as compared with the older type of poppet or with the newer type of low-lift plate valve, is the fact that it seats not by impact of the entire valve, but by increasing contact from the ends to the center. This characteristic, in combination with extreme lightness and flexibility, not only results in the marked durability demonstrated by the test carried out, but also permits a lift area greatly in excess of that obtainable with the annular low-lift type, seating by impact. The valve as applied to a compressor is shown in Fig. 4.

In efficiency of performance the valve shows actual measured volumetric efficiency approaching within less

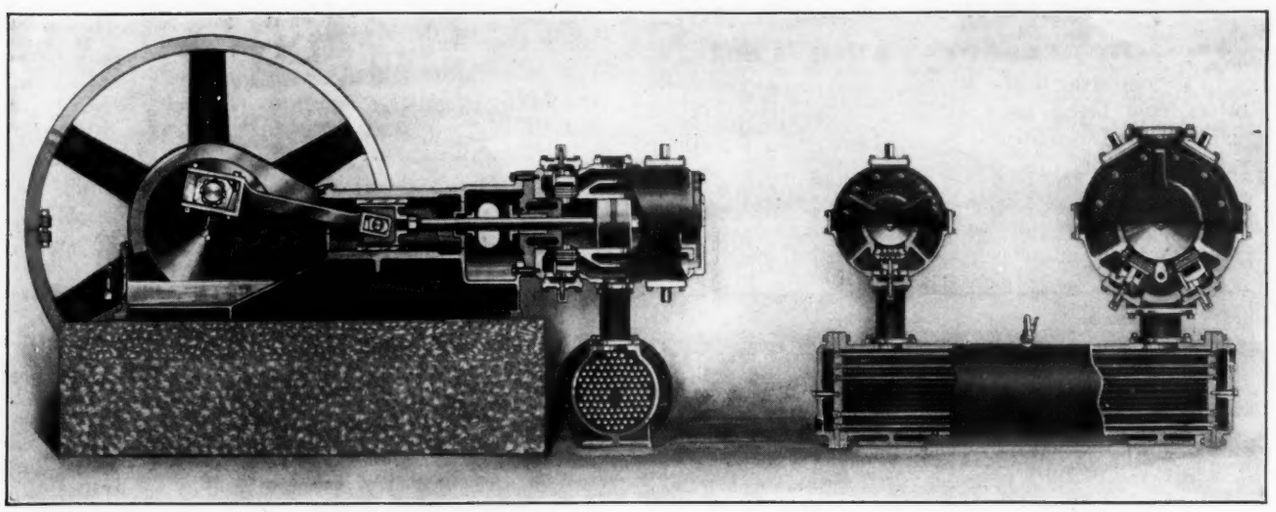


FIG. 4. FEATHER VALVES APPLIED TO AN AIR COMPRESSOR

than 1 per cent. of the efficiency indicated by a diagram, which latter indicated efficiency shows only the reëxpansion loss from clearance. This close approach of the actual to the indicated efficiency is a significant check on the valve performance, inasmuch as the clearance reëxpansion indicated by the card, and which is inevitable in any practicable air compressor, is not an economical loss, the energy return of the clearance reëxpansion being practically identical with the energy absorbed in its compression.

Contact seating, in addition to contributing to a high degree of durability and permitting the high lift which gives the valve its exceptional efficiency, also accounts for its remarkably quiet action.

Extreme simplicity of makeup permits the valve to be made reversible as regards seat and guard, so that the same valve can be used in its cylinder optionally as an intake or a discharge valve, the construction of both being identical and their function being determined only by the relation of the seat to the cylinder bore.

## Nugent's Indicator Reducing Motion

A simple indicator reducing motion has been designed recently by Wm. W. Nugent & Co., of Chicago. It may be attached to any Nugent telescopic crosshead-pin oiling device or to a special stand on the floor. The former

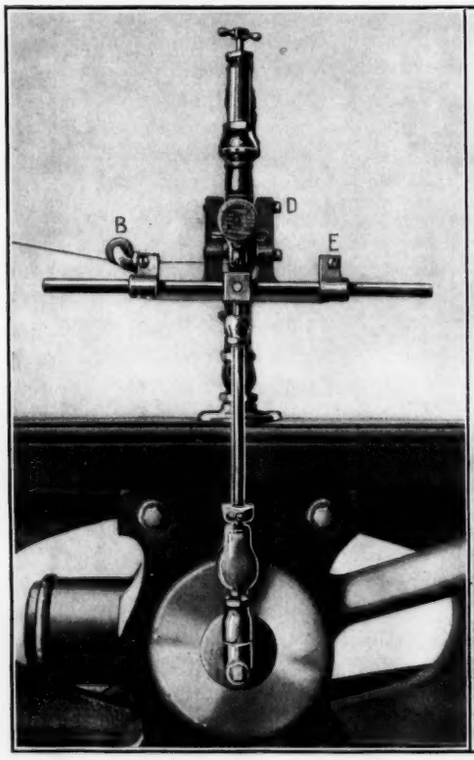



FIG. 1. NUGENT REDUCING MOTION ATTACHED TO CROSSHEAD-PIN OILING DEVICE

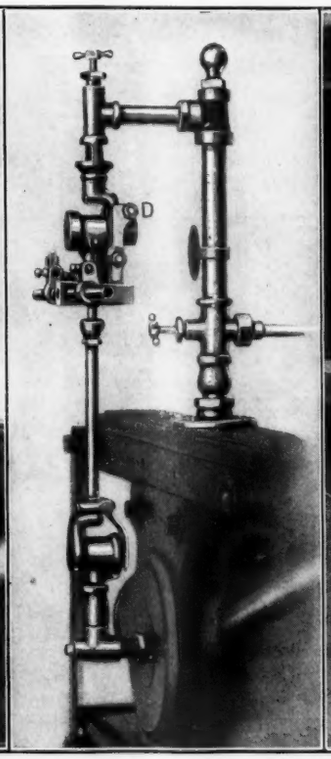



FIG. 2. SIDE VIEW OF REDUCING MOTION AND OILING DEVICE

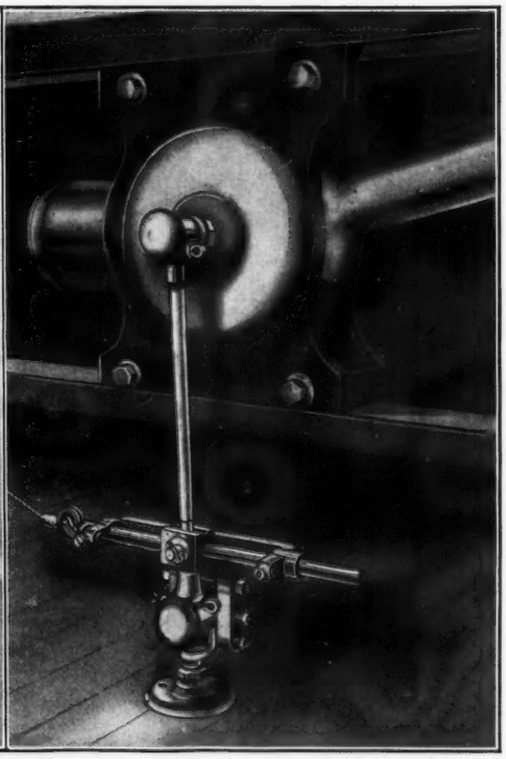


FIG. 3. REDUCING MOTION SUPPORTED BY STAND SECURED TO THE FLOOR

method of attachment is shown in Figs. 1 and 2 and the floor stand in Fig. 3. Either is mounted at the center of the stroke.

As shown in the first two illustrations, the reducing motion consists of a forked member which straddles and is clamped to the oiling device mounted on the top guide to the crosshead. The forked member just referred to supports two horizontal tubular guides for an oscillating block which reproduces on a smaller scale the motion of the piston. The block is moved back and forth by the telescopic tube of the oiling device. A pin that is free to turn passes horizontally through the block. The former is bored radially to allow the telescopic tube to pass through.

A given point on the tube naturally moves in the arc of a circle, but the arrangement allows the block to travel horizontally, as it may slide along the tube away from or toward the fulcrum, as it moves either side of the central position. As the pin is free to turn, there is no binding action on the tube. The cord is attached to this same pin and on its way to the indicator passes through a swivel guide pulley. The device accurately reproduces the motion of the piston and can be easily attached to or removed from the supporting element.

## A Question Puzzling to Some

Many of our very practical engineers do not understand the reason for the greater efficiency or economy of a compound condensing over a simple condensing engine. The prevailing idea seems to be that power and efficiency are derived only from a greater number of cylinders. The principal difference in the two engines is entirely overlooked—the difference in the loss of heat through cylinder condensation; otherwise the greater number of cylinders would be of no advantage, but a disadvantage.

Consider the diagram of a 300-hp. simple condensing engine, admitting steam at a pressure of 200 lb. absolute and a temperature of 287.2 deg. F. and expanding down to 1 lb. absolute and 102.9 deg. F., thus losing 284.4 deg. F.; or, in other words, admitting steam at the high temperature of 387.3 deg. F. into a cylinder which has been cooled to some extent by the expanded steam at 102.9 deg. F., thus condensing and losing a large percentage of the incoming steam in reheating the cylinder.

With a compound engine with steam at the same pressure and temperature, and the high-pressure cylinder expanding it down to 27 lb. absolute at 245.1 deg. F., the temperature difference between the admission and exhaust is only 142.2 deg. F. This exhaust from the high-pressure cylinder is admitted into the low-pressure cylinder at practically the same pressure and temperature, and expanding in its turn to 1 lb. absolute and 102.9 deg. This makes a difference of 142.2 deg., the same difference of temperature between the admission and exhaust in the high- and low-pressure cylinders.

By having two cylinders, as in the compound engine, the extreme difference in temperature met with at admission in each cylinder is reduced.

**Rate of Combustion** is the amount of fuel burned per hour per square foot of grate surface. It varies from about 5 lb. in small furnaces to 100 in large furnaces under forced draft. The ordinary rate for anthracite is from 5 to 15 lb. and for bituminous coal 5 to 25 lb.; in locomotives, from 45 to 90 lb.

## Just for Fun

A foreigner who spoke very little English was second fireman in a plant of eight 150-hp. horizontal return-tubular boilers. The so called "Hunkie" was told one day to take one of the boilers out of service for cleaning and repairs. He shut the header valve all right, and when the steam pressure had fallen to 60 lb. he went to blow down the boiler, but the blowoff pipe was completely stopped up. He then took the blind flange off the cross on the end of the pipe and commenced poking at the obstruction with a rod while there was still 60 lb. pressure on the boiler. Somebody caught him before he completed the "suicide act."—*F. F. Jorgensen, Gillespie, Ill.*

---

In a plant where I was employed a man who was interested in the plant and was supposed to understand electricity came in one evening during the peak-load period. There were two machines, and No. 1 was permanently connected up, but No. 2 was only temporarily connected, and in such a manner that one main circuit-breaker served both machines, although only one could be run at a time. The feeder switches were all of the quick-break type. These switches were something new to the gentleman, and I showed him how they worked, and remarked that the large one was the main switch for the idle machine. At this time I turned to a storage-battery panel, and bang went the circuit-breaker and everything went out. He had simply tried that large quick-break switch, but when things were in order again he remarked: "I guess it is a good idea to leave things alone that we are not acquainted with."

In this same plant we were required to take ground readings from all of the circuits once a day, before the evening load came on. The "boss" had a habit of coming in frequently and pulling out and putting in switches and watching the effect on the ground indicator. On one occasion he came in before I had tried the circuits and proceeded as usual. It happened that I went over to the engine while he was engaged with the switches and coincident with my move he threw in a switch which had been open, and started a commotion. I was innocent of any connection with the short-circuit, which was on that line, but circumstantial evidence was against me. Our "friendly relations" were not "strained" at all, but the "boss" sort of lost interest in grounds.—*H. L. Strong, Yarmouthville, Me.*

---

The fellow who painted the commutator has nothing on a painting stunt pulled off by one of my men last fall. I decided to give the machinery a coat of paint. The boys prided themselves on the appearance of the lagging, valve-gear and especially the well polished cylinder head of a Corliss engine. One afternoon all the painting was completed with the exception of an oil guard on this engine. The middle-watch man was told not to run the machine that night, but to finish the painting job.

The following morning I found our nice lagging, cylinder head, bonnets and valve-gear, including the sole plate and dashpots, all painted green, and it had begun to run and then baked. The color of the atmosphere was "some variegated" while we were scraping paint.—*E. B. Mertens, Milwaukee, Wis.*

# Automatic Electric Elevator Dispatcher

By Norman G. Meade

Dispatching of passenger elevators in large office buildings has received considerable attention recently, and it has been the aim to obviate the human element as far as possible. The engineering staff of the Insurance Exchange Building, Chicago, has designed and installed an automatic electric dispatcher which times the starting of the elevators from both the bottom and the top floors. This building is a modern 18-story office structure covering half a city block and is equipped with sixteen electric passenger elevators arranged in four banks on four sides of the building.

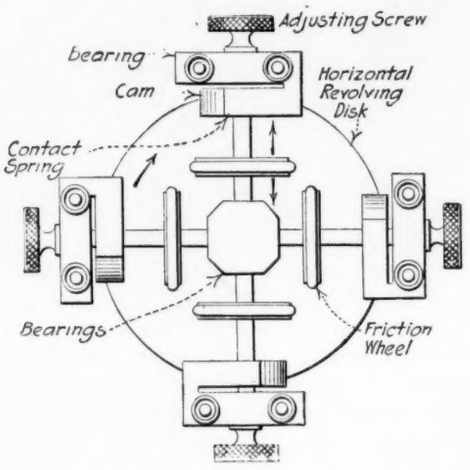



Fig. 1. Plan of Dispatcher

These elevators are of the Otis traction type, operated by 25-hp., 220-volt direct-current motors controlled by Otis type MF4 controllers. Eight operate as express cars and eight as locals. The floor signals for the former normally do not operate below the twelfth floor, but on occasions such as Sundays and holidays, a switching arrangement is provided which throws in the signals for all floors. The automatic dispatcher presents several novel features and signals the starting of the elevators independently of the hall men, who merely direct persons to their desired destination in the building.

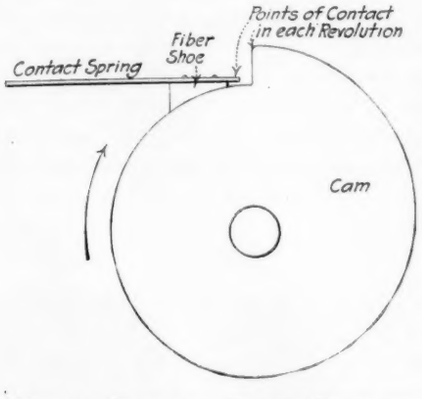



Fig. 2. Cam and Spring Contact

There is a bell located at the top and at the bottom of each bank of elevators, operated by the dispatcher. This bell is adjustable so that the timing of the cars may range from 20 to 60 sec. A 220-volt to 15-volt motor-generator set, with an auxiliary storage battery, furnishes energy for the dispatcher.

Fig. 1 is a plan view of the dispatcher, which consists essentially of a horizontal revolving disk operated through reducing gears by a 1/6-hp. motor at a constant speed. Four vertical columns attached to the slate base support four independent hollow shafts to which are attached friction wheels and cams; a contact spring rests on each cam, but is normally out of electrical contact except once in each revolution. The horizontal shafts are hollow and in-

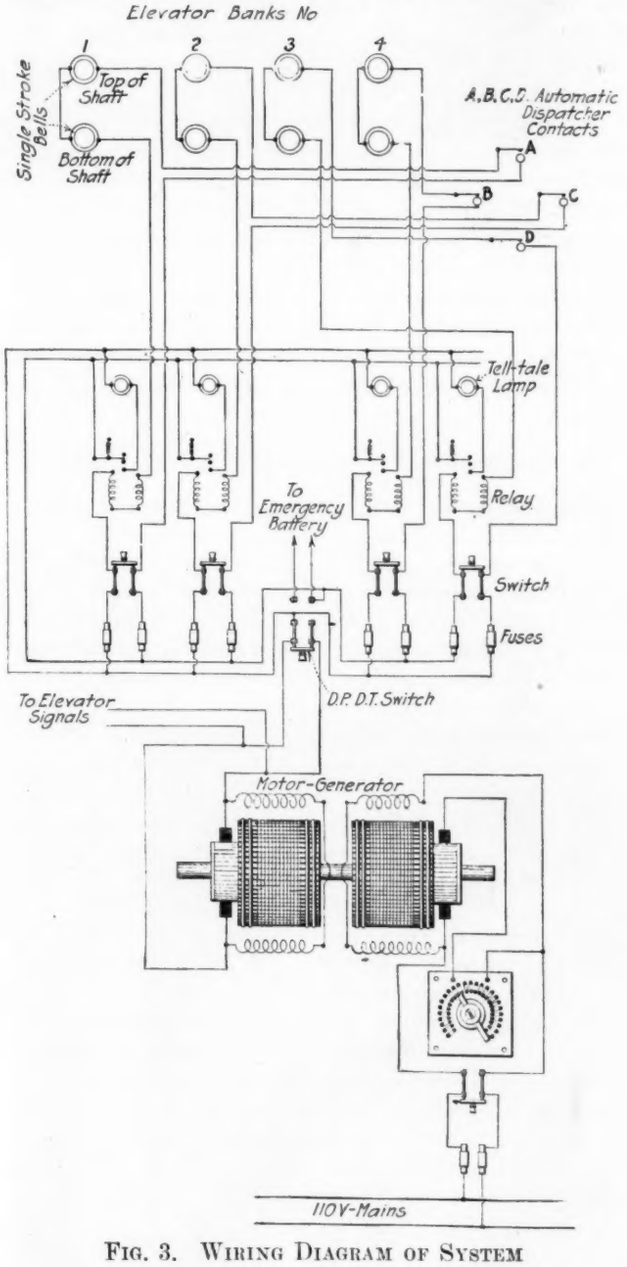



Fig. 3. Wiring Diagram of System

close a threaded rod which engages with a central projection from the friction wheel that passes through a slot in the shaft. Hence, the position of the friction wheel on the shaft can be varied by turning the adjusting screw one way or the other. Moving the friction wheel toward the center of the revolving disk decreases the speed and moving it toward the outer edge increases it.

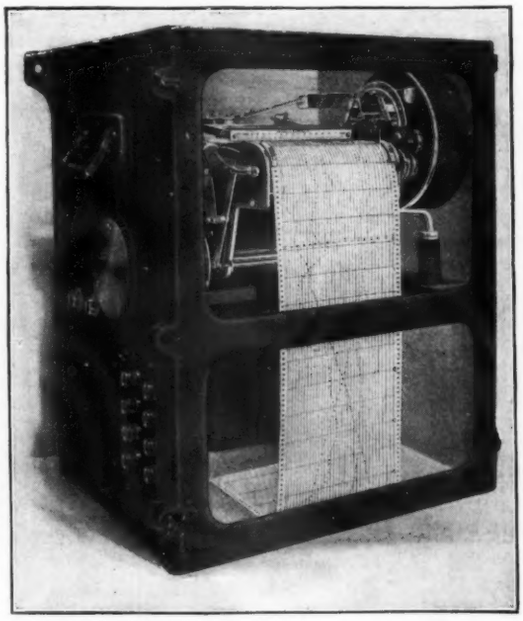
Details of the cam and spring-contact arrangement are shown in Fig. 2. The former is insulated from the latter by a fiber shoe, except when the spring escapes over the

lip of the cam, and an open circuit normally exists. Fig. 3 is a wiring diagram of the motor-generator set and the dispatcher with its switchboard. The double-pole, double-throw switch in the center of the board is for disconnecting the motor-generator set and connecting the batteries in case of emergency. A relay is connected in the generator circuit and in case of failure of the latter, releases its armature which closes a local circuit and lights a telltale lamp, warning the attendant.

The dispatcher equipment on the board consists f four sets of fuses, four double-pole switches, four telltale lamps, and four relays—one o tfit for each bank of elevators. As the horizontal shafts of the dispatcher revolve, contact is made each revolution as the contact springs slip over the lips of the cams, closing the circuit through the relay coils and the single-stroke bells which are connected in series. The relay draws down its armature and closes the circuit through the telltale lamp, which lights, indicating that the signal has been given. Cars start in order in each bank of elevators, one after the other at the signal, and return from the top in the same order on receiving the signal.

## Recording Pyrometer

The recording pyrometer shown is of a new type developed by the Wilson-Maeulen Co., 1 East Forty-Second St., New York City, and is to be marketed under the trade name of "Tapalog."



"TAPALOG" AUTOGRAPHIC PYROMETER

A special feature of this instrument is that the carriage containing the record paper, typewriter ribbon and other recording mechanism is pivoted to drop away from the galvanometer so that the paper and ribbon can be changed without danger of injuring the galvanometer.

The record is taken on the under side of the strip. The tracing paper used is visible from both sides and the speed of the record is 1 in. an hour. A dot is made every 12 seconds, and during the intervening period the pointer is free to swing to its true position.

The depressing member which makes the dot is operated by a three-cell dry battery; the clock merely shows the time. The indicating scale is mounted on the front of a chopper bar that is pulled down by an electromagnet, with a blow to make the record, and this bar is over-counterbalanced so that it tends to rise quickly after the downward stroke, at the bottom of which the electromagnetic circuit is opened.

The Tapalog will take a single record of one temperature in one color, but it is generally furnished with an automatic switch which, every minute and a half, switches the Tapalog from one thermocouple to the next and brings another portion of the multi-color typewriter ribbon under the pointer, so that the different records are taken in as many as four distinctive colors.

Multiple recording is important, for instance, in a large furnace where thermocouples are inserted at different points; the records, all being on one sheet, show definitely whether the furnace is evenly heated or just what the degree of uneven heating may be—the whole graphically set forth in a multiple record. The ribbon that passes under the record paper is made in the form of an endless belt so that it is not necessary to employ any mechanism to reverse the direction of travel.

This pyrometer may be located a long distance from the furnace, and to overcome shrinkage it is provided throughout with bakelite disks, washers, plates and other working parts.

## Steam-Turbine Rolling-Mill Drive

The rolling mill has been one of the last stands of the large reciprocating engine. The Carpenter Steel Co., of Reading, Penn., has, however, recently installed a low-pressure steam turbine for driving two stands of 18-in. three-high mills. The turbine is of the De Laval multi-stage impulse type, in nine stages. It runs at 5000 r.p.m., and the speed is reduced by double helical involute gears, first to 600 and then to 100 r.p.m. The Jahns governor, with which it is supplied, may be adjusted while the machine is in operation, to vary the speed from 100 to 70 r.p.m. on the mill shaft.

The turbine will operate normally with 3 lb. gage pressure at the turbine throttle, and with 3 in. absolute pressure in the turbine wheel case at the exhaust end, under which conditions it will carry 350 hp. at the speeds mentioned and require not more than 26 lb. of steam per hour per brake horsepower, the power being measured at the end of the second gear reduction. It may be run low-pressure condensing, mixed-pressure condensing, high-pressure condensing, and high-pressure noncondensing.

With 120 lb. pressure at the throttle and 3 in. absolute pressure in the turbine wheel case, it is guaranteed not to require more than 17½ lb. of steam per brake horsepower-hour. As a mixed-pressure turbine, with the same pressure conditions, it will carry 600 b.hp. continuously, and is also designed to carry the full load of 600 hp. on the high-pressure steam alone, under which condition it is guaranteed to take not more than 15.7 lb. of steam per brake horsepower-hour.

**American Cigar Co.'s Plant**—We have been informed by Clark, MacMullen & Riley, 101 Park Ave., New York City, that the plant of the American Cigar Co., Garfield, N. J., described by W. L. Durand in Apr. 6 issue was designed and installed by them. This building is the first in the country used exclusively for cigar manufacturing to put in an air-washing system.

## Vacuum Fluid Cooler

This device has been designed for the purpose of reducing the temperature of water below that of the surrounding atmosphere. Referring to Fig. 1, where the cooler is shown in the form of a rectangular tower, the air is taken through the air-cooling chamber, details of which are shown in Fig. 2, at a velocity of about 2000 ft. per min. In this chamber is a manifold sprayer which subdivides the water after it has passed through a strainer, Fig. 3. As the air enters through the cooling chamber it is compressed by passing through a smaller orifice at a high velocity, and is thereby reduced to about 65 deg., at which temperature it is drawn through the cooler and comes in direct contact with the falling water. The method of baffling the water and the air by means of tongued baffle plates, Fig. 4, causes the drops of water to come in contact with the cooler air twelve or fourteen times during their fall from the inlet to the outlet.

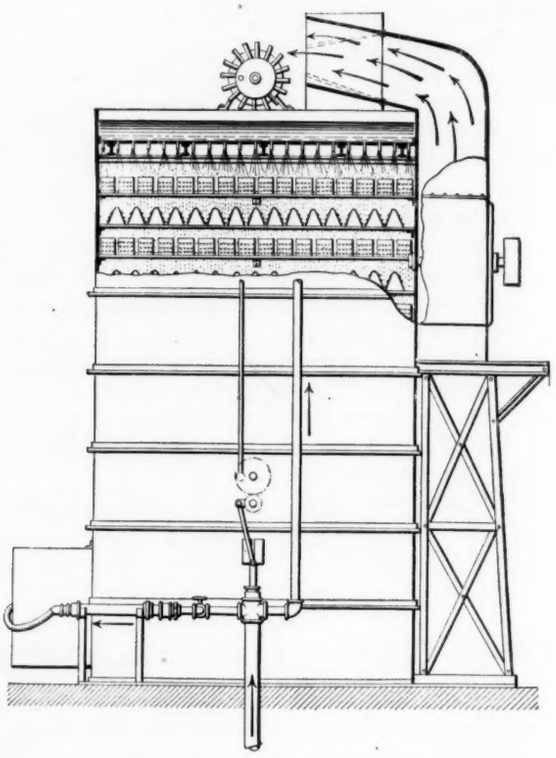
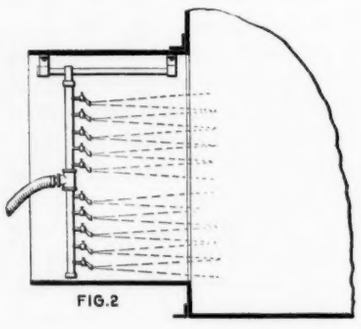
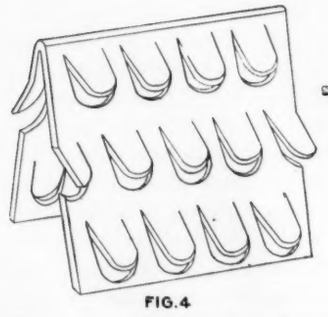


FIG. 1. COOLING TOWER



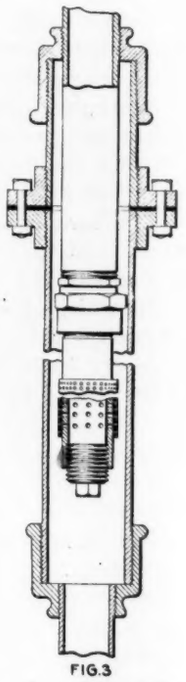




FIG. 2. AIR-COOLING CHAMBER. FIG. 3. STRAINER. FIG. 4. TONGUED BAFFLE PLATES

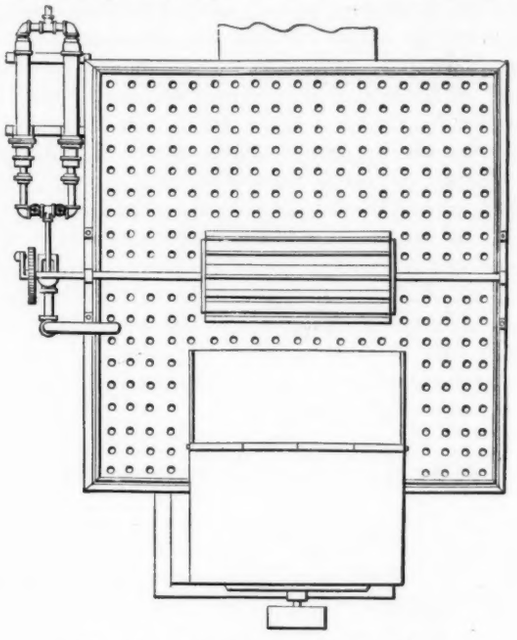


FIG. 5. PERFORATED STEEL PAN

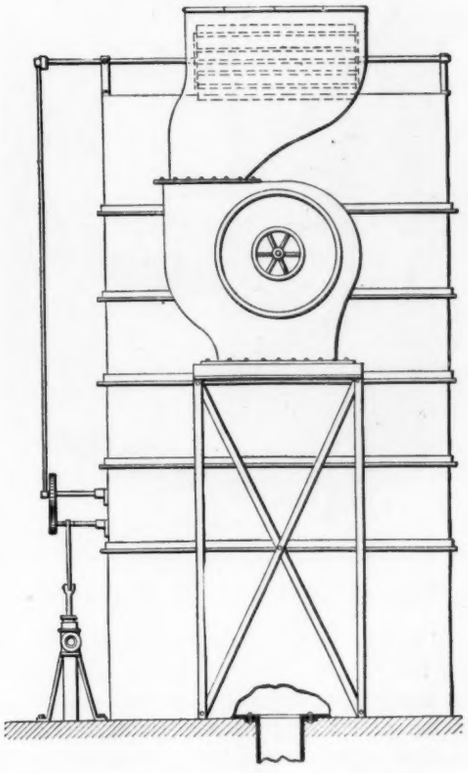


FIG. 6. END VIEW OF COOLING TOWER

At the top of the cooler is a steel pan about twenty inches deep (Fig. 5), into which the hot water from the condenser is pumped and the bottom of which is perfo-

rated at regular intervals with 1-in. holes. Each hole is connected to a funnel-shaped tube, Fig. 4, which tapers to ¼ in. diameter at the base, thus forming sprayers. These tubes are so designed that with the weight of water which passes through it is impossible for them to become clogged or to retain any foreign matter that may be in the water.

The water is distributed to the tubes in the form of a spray, and when sufficient is in the pan to cover the holes, the exhaust fan, Fig. 6, placed under the tube, forms a partial vacuum into which is precipitated the falling hot water and vapor.

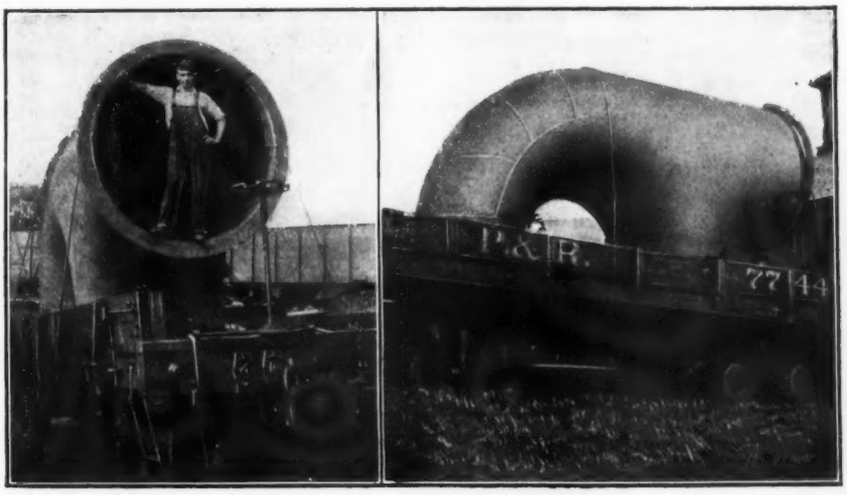
The exhaust is under the top pan. By circulating the water through the cooler after it has been heated from 90 to 160 deg. F. in passing through a condenser, it is brought back to be used again at a temperature of 70 to 75 deg. F. after passing through the cooler.

This cooling tower is manufactured by the Vacuum Flue & Cooler Co., 205 Highland Bldg., South Highland Ave., Pittsburgh, Penn.

## Large Double Exhaust Fitting

The accompanying illustration shows a 72-in., cast-iron, double exhaust fitting which was recently made in the shops of the Hunsicker Engineering Co., Lebanon, Penn. So far as known this is the largest fitting of its kind ever made in the East.

It is used in connection with a 5000-kv.-a. turbine, 1500 r.p.m., at 80 lb. boiler pressure. The condenser used in the equipment is a No. 23 barometric type. The double outlets connect with the expansion joints of the turbine and the large end connects with a 72-in. exhaust line. It weighs 14 tons and is 18 ft. in length over all.




TWO VIEWS OF 72-IN. DOUBLE EXHAUST FITTING

**The Efficiency of Compressed Air** can be greatly increased by reheating. The gains are both direct and indirect, the chief direct gain being in the greatly increased efficiency of fuel used in the heating stoves as compared with the effect when coal is burned under boilers. It is commonly stated, and the statement is fairly correct, that when 1 lb. of coal is burned in a reheater stove the commercial effect is as great as when 3 lb. are burned under a boiler. The increase in commercial efficiency when reheating air from 60 deg. F. to 400 deg. F. may be put at 35 per cent. The indirect gains are better lubrication of the compressed-air engine, less investment required, as a smaller plant will be needed, reduction of compressor-engine friction as compared with the useful work done.—Exchange.

## Belting Calculations

BY E. O. WATERS

The theory of belting, as presented in standard handbooks, does not seem to harmonize with examples taken from everyday practice. There appears to be conflict between textbook theory and machine-shop practice, but this conflict is more apparent than real.

Primarily, a belt is made in such a length that it will be stretched enough to give it an initial tension. When the driving pulley starts it will cause one-half of the belt to tighten and the other half to become more or less slack. When the difference in tensions becomes slightly greater than the resistance of the driven pulley, it, in turn, will start up, provided the resistance does not exceed the maximum value determined by the coefficient of friction between the belt and pulley surface and the arc of contact made by the belt. This maximum ratio is that number whose common logarithm is $0.00758\mu\theta$, where $\mu$ is the coefficient of friction and $\theta$ is the arc of contact measured in degrees. For any proposed installation of belting $\theta$ is known (it is usually about 180°); and $\mu$ was determined many years ago by hanging pieces of belting with weights attached to the ends, over fixed sheaves, and finding out what excess weight hanging from one end was required to produce slipping. Accordingly, our maximum limit for $\frac{T_t}{T_s}$ is fixed, and since $T_t - T_s$ is determined by the power to be transmitted and the belt speed, we have only to make $T_t$ and $T_s$ large enough so that their ratio will be within the required limit, and to choose a belt sufficiently wide and thick to stand this maximum pull.

Suppose, for example, that it is desired to transmit 100 hp. between two 30-in. cast-iron pulleys, 30 ft. apart and on the same level, at a belt speed of 2500 ft. per minute. The resistance of the driven pulley will be 1320 lb. at the rim, and this same figure will represent the excess of $T_t$ over $T_s$. Since the arc of contact is 180 deg., and $\mu$, according to the experiments referred to above, is about 0.3, $\frac{T_t}{T_s}$ must be greater than 2161 lb. and $T_s$ greater than 841 lb. Of course, the excess will be made as small as possible in order to avoid undue pressure on the shaft bearings and an unnecessarily large belt; but there must be an actual excess, or the belt will soon stretch permanently—so much so that it will be necessary to shorten it by cutting out a piece, so as to keep it from slipping. As for the size of belt to be used, the standard rule of 111 lb. maximum tension per inch width of double belt, corresponding to a maximum tension of about 375 lb. per sq.in., gives us a belt having 5.75 sq.in. sectional area, which would probably mean a double belt 20 in. wide. According to F. W. Taylor's recommendations, the maximum tension should be only 54 lb. per inch width of double belt, if it is desired to have belts which will give the least expense for repairs and lost time on machinery. In that case, our calculations show that a 30-in. double belt or, preferably, a 20-in. quadruple belt must be used.

So much for the design of belting transmissions by theoretical formulas. On the other side of the question, our own observation tells us, first, that for transmitting 100 hp. at 2500 ft. per min., very few factories use belts much over 20 in. wide and ⅜ in. thick, and second, that such belts, when running between pulleys a considerable distance apart, will often sag on the loose side anywhere from one-third to three-quarters of the mean diameter of the pulleys. Now we can very easily find the tension at the two ends of the loose portion of the belt by means of a little mechanics and the use of the well-known fact that a belt or rope, when suspended between two points, hangs approximately in a parabolic curve whose equation is

$$y = \frac{wx^2}{2H}$$

in which

$y$ = Vertical distance of any part of the belt from the lowest point;
$x$ = Horizontal distance of the same point from the lowest point;
$w$ = Weight per unit length, and
$H$ = Tension at the lowest point.

Taking $y = 2$ ft. and $x = 15$ ft., corresponding to a point on the loose side of the belt just as it is leaving one of the pulleys, and assuming the weight of leather as 0.036 lb. per cu.in., we have, for the 20-in. double belt, $w = 2.47$ lb. per ft. and $H = 139$ lb. At one end of the span the tension $T_s$ will consist of a horizontal component equal to 139 lb. and a vertical component equal to the weight of half the span, or approximately 37 lb. This gives us $T_s = 143.8$ lb., which, to say the least, does not check very well with the minimum value of 841 lb. obtained by theoretical deductions. Even if the heavier belt were used, according to Taylor's recommendations, $w$ and $H$ would merely be doubled, and $T_s$ would then be 288 lb.

In the opinion of the writer this discrepancy is due to two things—the arc of contact and the coefficient of friction assumed in the theoretical formula $\log \frac{T_t}{T_s} = 0.00758\mu\theta$. The arc is increased very materially by a slight sag in the loose side of the belt, if that side is the upper one. In our problem a sag of 2 ft. at the middle of the span means an increase of about 15 deg. in the arc of contact, and this difference alone is enough to increase the allowable ratio of $\frac{T_t}{T_s}$ from 2.57 to 2.78. However, a much greater change is caused by the second factor. It is not generally known that the coefficient of friction is very decidedly increased by slippage between the belt and pulley surfaces, although this fact seems to have been recognized many years ago by several investigators of the practical mechanics of belting. Unwin gives the equation $\mu = 0.2 + 0.004\sqrt{V}$, where $V$ is the belt velocity in feet per minute, on the assumption that the amount of slip will increase as the speed of a belt becomes greater. Barth, after making a very careful analysis of some tests carried out by Wilfred Lewis for Wm. Sellers & Co., deduced the two following equations: $\mu = 0.6 - \frac{2}{4 + v}$ where $v$ is the velocity of slippage of the belt relative to one of the pulleys, in feet per minute, and in the other $\mu = 0.54 - \frac{140}{500 + V}$.

The last of these, like Unwin's equation, depends only on the velocity of the belt, and is supposed to be used for ordinary belt transmissions where the total slip between the driving and driven pulleys is about 1½ per cent. Now a slip of 3 per cent. is very common in belting practice, and for belts that are running at all loose it may easily reach 6 per cent. In such a case, a belt running at 2500 ft. per min. will slip over each pulley with a velocity of 75 ft. per min., and $\mu$, according to Barth's first formula, = 0.575. At that rate, the ratio $\frac{T_t}{T_s}$, as figured by the same equation that gave us $\frac{T_t}{T_s} = 2.57$, may be increased to 7.1, and $T_t$ and $T_s$, instead of being, respectively, 2161 lb. and 841 lb., are reduced to 1536 lb. and 216 lb. This value of the tension in the loose side of the belt lies about half way between the tension which it was figured would be required to support a 20-in. double belt with a 2-ft. sag between the pulleys, and that required for a 20-in. quadruple or 40-in. double belt with the same sag. In other words, a 20-in. triple belt can be used in the transmission which we have been figuring, and allowed to sag 2 ft. on the loose side, in spite of the preliminary calculations which showed that such a heavy initial tension would be needed that the belt would pass from pulley to pulley in practically straight lines.

The upshot of the whole matter seems to be this: Almost any value within reason for the ratio $\frac{T_t}{T_s}$ may be obtained by calculation, simply by juggling with the coefficient of friction; it is therefore worse than useless to use the standard formulas for the design of belt transmissions, unless we take into account such factors as the belt's speed and slip, which are usually neglected. If, however, we can get a pretty accurate value for the coefficient of friction, we can figure belt tensions which will agree quite satisfactorily with those which we observe in practice. It should be noted that this can be done without in any way taking into account the effects of centrifugal force, which, in the opinion of some, should be given the entire credit for making it possible to run slack belts.

## Economy of Heating Feed Water

By C. E. Anderson

Notwithstanding the economy of using exhaust steam for heating water for boiler feed and other purposes and the widespread information on the subject, the writer is constantly coming in contact with cases where live steam is used for heating water and for other purposes where exhaust steam might readily be employed.

Theory and practice prove beyond a doubt that a saving of 10 to 20 per cent. in coal consumption may be effected by heating feed water, the amount varying according to conditions. Where large quantities of hot water are used for other purposes a much greater saving may be made by using exhaust instead of live steam to heat it. In some cases it is more economical to run noncondensing, but if the exhaust furnished by the pumps and other auxiliary machinery is nearly enough, it is good practice to make up the deficiency by bleeding the intermediate receiver.

The following calculations may be of interest, showing the value of the feed-water heater. For convenience, assume a heater working continuously at its full capacity for ten hours a day and supplied with exhaust steam from the average noncondensing engine developing about 75 hp. Taking water at 60 deg. F., the heater will deliver about 1200 gal. per hour, or 12,000 gal. per day at about 210 deg. F. One gallon of water weighs about 8⅓ lb., and 12,000 gal. will weigh 100,000 lb. Since it requires 150 heat units (B.t.u.) to heat one pound of water from 60 deg. F. to 210 deg. F., 100,000 lb. requires 15,000,000 heat units.

Roughly speaking, one pound (weight) of live steam will furnish 1000 heat units, therefore, 15,000 lb. of live steam would be required to do the work. Under good conditions one pound of coal will evaporate 10 lb. of water. This figure may be exceeded in some instances, but probably the majority of cases fall below it. On this basis no less than 1500 lb. of coal per day will be required. Assuming a working year of 300 days, this gives a total of 450,000 lb., or 225 tons, of coal burned, which at $4 per ton would cost $900.

This sum represents the saving by the use of a heater with exhaust steam. Of course, these figures are arbitrary and the result will be more or less fully realized in practice, according to circumstances.

## Curnon Steam Meter

The Curnon Steam Meter is nothing more than the familiar low-pressure recording gage of the diaphragm type, and its pressure regulator which utilizes the Bourdon tube. The meter is sensitive and fairly accurate. The charts used with the instrument are directly calibrated in pounds of steam per hour, so that no calculations are necessary to secure the desired data. The installation is simplified by the use of double cocks both on the test plug and on the meter.

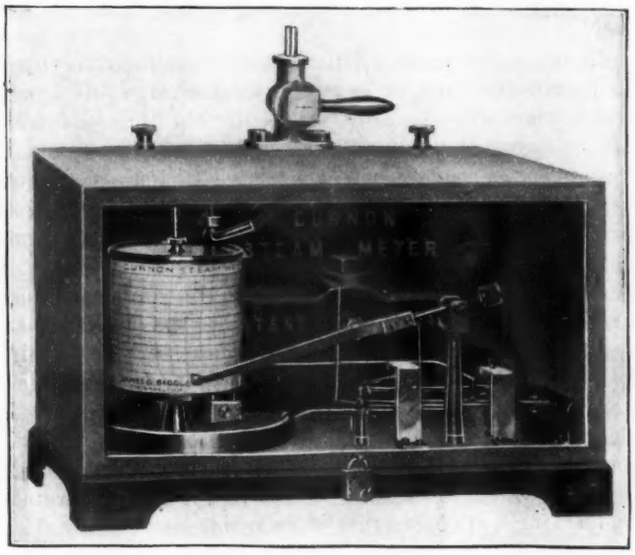


Fig. 1. Curnon Steam Meter

The principle employed in this meter, Fig. 1, is that of the Pitot tube, consisting of two small tubes with their ends bent at right angles and inserted in the pipe in which the steam is flowing in such manner that the end of one faces against, and that of the other with, the direction of flow. The rush of steam past the tubes causes in one a slight increase, and in the other a corresponding decrease, in the static pressure in the pipe. The difference in pressure is a measure of the velocity of the steam, and by connecting the two tubes to a sensitive differential pressure recorder a record is obtained of the quantity of steam passing through the pipe to which it is attached. Fig. 2 shows the Pitot tube mounted in the Curnon plug. The special feature is that the tubes are of such length

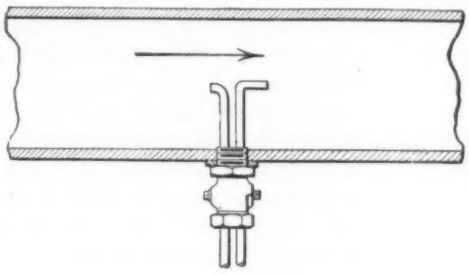


Fig. 2. Arrangement of the Pitot Tube

as to reach exactly to the center of the pipe, where the speed of the flow is highest and where eddy currents are least likely to interfere with the accuracy of the measurements. The low-pressure tube has a long elbow, which further increases the pressure difference and the motive power available to secure clear records of small variation in load. A single three-way cross controls both tubes and enables steam to be blown through them for cleaning purposes.

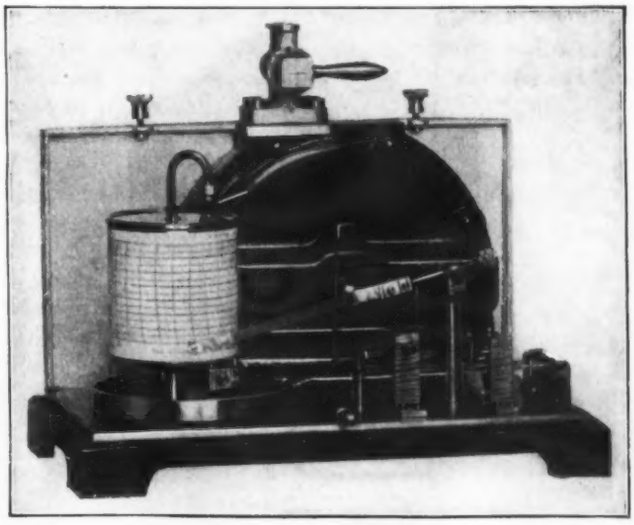


Fig. 3. Meter without Casing

All of the moving parts are mounted on a cast-iron base, as shown in Fig. 1. The outside dimensions when mounted in a box are 15x9x9 in. and the weight is about fifty pounds. Fig. 4 is a view of the diaphragm box, which is arranged in the back plate of the meter case. The two pressures in the test plug act upon the sides of a sensitive diaphragm, which is stiffened by two plates and controlled by a strong flat spring. The movements of the diaphragm are conveyed to the pen through a system of levers which multiply and rectify the movement in such a way that the curves traced upon the paper chart directly represent the weight of steam passing through the pipes.

As the accuracy of the instrument would be affected by any variation in boiler pressure, each is fitted with an automatic pressure regulator consisting mainly of a hollow spring of the Bourdon type, arranged centrally below the recording drum and linked at its free end to the multiplying gear. This is shown in Figs. 1 and 3. The interior of the Bourdon tube is in communication with the steam pressure in the pipe, and the movement of the free end so acts upon the pin gear as to automatically correct the reading for any fluctuation in pressure.

The connection between the meter and the ⅜-in. copper test tube may be any reasonable distance away, either above or below the pipe.

The double cock, shown in Figs. 1 and 3, at the top, is constructed so that both ports are opened and closed simultaneously. In the closed position the meter is connected to the atmosphere and the pen must always stand at zero on the chart. This enables the operator to check the adjustments at any time. Between the plug and the meter are the condenser coils, which consist of ⅜-in. copper tubing fitted to the plug horizontally and on an exact level with the Pitot tube in the steam space. The

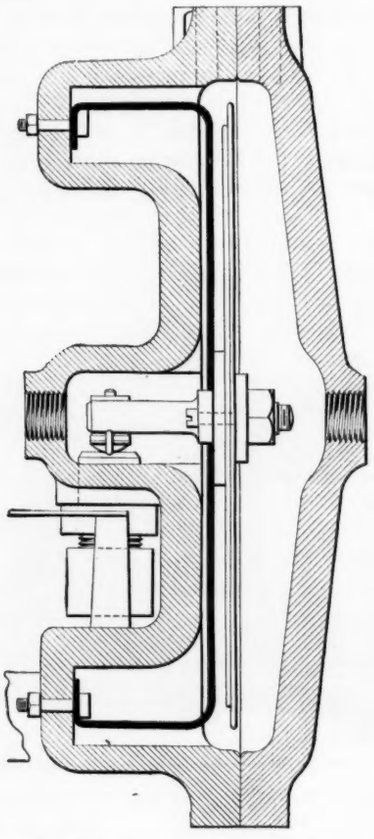


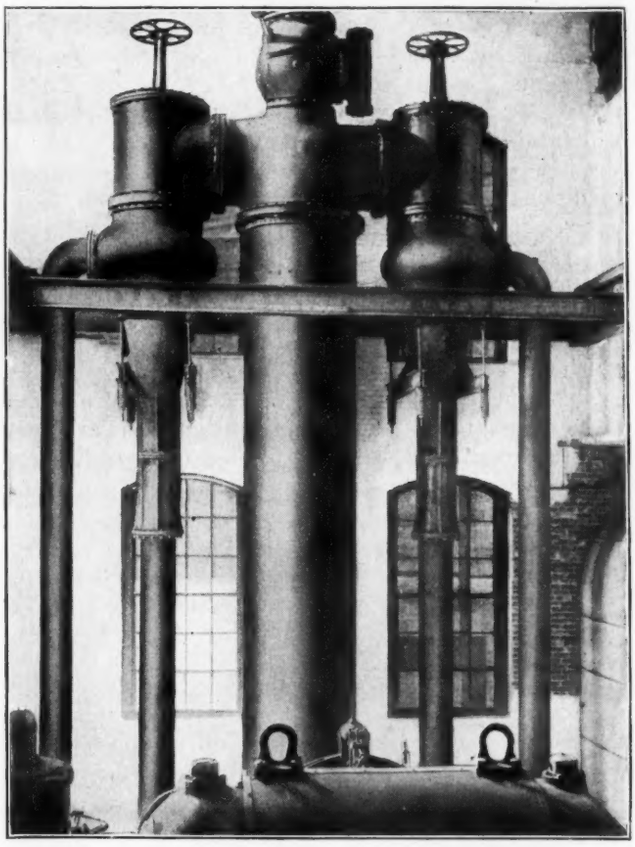
FIG. 4. DIAPHRAGM BOX IN SECTION

condensing coils insure that the meter, as well as the connecting pipe, is always filled with comparatively cold water, regardless of the displacement due to the movement of the diaphragm.

The meter is manufactured by the James Biddle Co., 1211 Arch St., Philadelphia, Penn. Each one is supplied with a test book, condenser coil, two 10-ft. copper connecting tubes and 100 charts and ink. The standard charts are calibrated in pounds of steam power and are available in various ranges for each size of pipe. The instrument is fitted with a 24-hr. chart and an 8-day clock.

## Deane Ejector Condenser

The illustration shows a pair of 34-in. ejector condensers recently installed by the Deane Steam Pump Co. in the city electric generating plant of Holyoke, Mass. The apparatus serves a 3500-kw. horizontal double-flow turbine and maintains 28 to 28½ in. of vacuum in the turbine exhaust chamber.



A PAIR OF 34-IN. EJECTOR CONDENSERS

In the construction a lifting device has been provided to permit the cone to be raised to discharge any trash which may stop the annular opening in the condenser after passing through the strainer. The tees shown at the top of the condenser are so designed that the cone can be lifted through the tee in case of desired changes, repairs or alterations. A pointer on the cone-raising spindle indicates the amount of opening around the cone. A device has also been incorporated to center the cone and prevent vibration of the lower edge when it is in its working position. The long run of 50-in. exhaust-steam riser and the 16-in. injection pipes are noticeable. These are both made of lap-welded steel tubing. The exhaust-steam riser is in one piece 25 ft. 8¾ in. long and the injection risers and the drop pipes are also each in one piece. Expansion and contraction of the piping under varying temperatures are taken care of by spring suspensions.

**Formidable Looking Formulas**—Many engineers will go down without a struggle before a formula which has a logarithm, entropy or a sine, cosine or tangent in it. It is just as simple to look up one of these quantities and to substitute the value given in the table for the letters of the formula as it is to hunt up the steam temperature corresponding to a given pressure, or the area corresponding to a given diameter, and the same book which contains the tables of the properties of steam and of circumferences and areas will usually have the other things, too.

# Editorials

## Changes at Seventy-Fourth Street

There are few more forceful examples of what the economy of the large steam turbine means in the generation of power than the recent changes in the Seventy-Fourth Street station of the Interborough Rapid Transit Company, New York City, which are described in this issue. When it pays to discard to the junk pile engines of large power, nearly as good as new, to make room for turbo-generator units, it shows that designers have been accomplishing things. And yet, in doing so they undo their admirable work of but a short time ago.

These great engines should not go into oblivion without a word being recorded of their life's history, too much of which has perhaps already been forgotten. The installation in the Seventy-Fourth Street station in 1901, beside being one of the most notable, was virtually the last stand of the large reciprocating engine for electric generating units.

They were designed by that Connecticut farmer boy, Edwin Reynolds, who became one of the eminent figures in the steam-engineering history of the country. It is told that Mr. Reynolds was invited to New York to discuss with the Manhattan Railway Company's engineers the subject of selecting the type of engine for the Seventy-Fourth Street station. When he left the E. P. Allis works in Milwaukee, the question was unsettled. At Albany a telegram informed him that a committee representing the railway company would meet him at the Grand Central station. When he reached Harlem, the story goes, he had sketched on the back of a letter the design ultimately adopted, the sizes of cylinders and principal parts being indicated. And the problem was to crowd 12,000 horsepower in each engine, which had to go in a limited space.

It seems that Fate had decided that this daring stroke should be about the last, for the turbine was beginning to demonstrate its possibilities. The first important installation was in 1899, when some 400-kilowatt machines were put in the plant of the Westinghouse Air Brake Company. About a year later attention here and abroad was centered in the two 2000-kilowatt units of the Electric Light & Power Company, Hartford, Conn. This was the most notable installation in point of capacity and because the expansion was completed in one cylinder. Then came the announcement that several 5000-kilowatt units were being built for the combined station of the Metropolitan and District roads, London.

The fact was being established that the larger the capacity of the turbine the more favorable was its performance, and 1905 saw contracts closed for a 10,000-kilowatt Brown-Boveri-Parsons machine for the Westphalian Electricity Works, Essen, Germany. At this time the Brooklyn Heights Railroad Company got two 7500-kilowatt units, which were then the largest turbines in this country.

There next followed the first real tryout of the turbine on a large scale, when in 1905 about 10,000 kilowatts in turbines were run in competition with a nearly equal capacity of reciprocating engines in the L-Street station of the Boston Edison Company. The results of these competitive trials do not need to be told.

And all this truly remarkable advance had been made almost simultaneously with the installation of large reciprocating engines in such stations as the Seventy-Fourth Street of the Interborough Rapid Transit Company; Waterside, No. 2, of the New York Edison Company, new in 1902; and the Fifty-Ninth Street station of the Interborough Rapid Transit Company, new in 1904. This is the most conclusive evidence that the early development of the turbine was rapid beyond the expectations of the best engineering talent of the time.

## "Elbow Room" in Power-Station Design

If power-plant designers generally were experienced in station operation, more consideration would be given to adequate space for equipment and attendance. The temptation to cut down on space is great, to save in the amount of material required to inclose the plant and in the fixed charges per horsepower of capacity. This consideration limits stations built in country districts as well as those on valuable city land, and the operating staff frequently finds itself lacking in room.

Some of the difficulties of attempting to operate in a cramped location may well be considered as an offset to the demands of an initial layout extremely economical in structural material. The importance of adequate clearance will be conceded by any student of "safety first." Where bare conductors are to carry high voltage, the designer generally appreciates the element of clearance. It is fully as important in many low-voltage plants to provide adequate spacing between conductors and metal work and between lines and horizontal or vertical passages. The commercial value of the loads carried, even if there were no personal hazard in the crowding of live conductors into restricted spaces, often makes it inadvisable to take chances of interruptions. It should never be possible for an engineer, climbing an iron ladder and momentarily losing his grip, to swing outward so far as to touch the exposed contacts of an instrument transformer. Designers who will properly separate circuits carrying high voltages sometimes neglect to spend enough money on low-tension bus structures and switch cells to protect the service and the operator from mischances.

A thorough investigation of the relation between the labor cost of station service and the space requirements of the plants covered would be of much interest. Equipment arrangement is important, but accessibility and freedom to reach apparatus without traversing circuitous paths are also. Main units are rarely so crowded today in new stations as to hamper removing sections of

machinery, but in the placing of auxiliaries and in the arrangement of piping, valves and platforms, much is often to be desired. A construction engineer who rose to a high place in a consulting organization recently remarked that few things in his early experience had proved of greater value than several months of service as a substation operator after the close of each day's work in the drafting department. The actual handling of the equipment gave him a knowledge of space requirements that was reflected in his later work. He knew what room the switchboard operator needed to manipulate manually connected switch levers in critical moments; he appreciated the desirability of being able to read instruments easily from all apparatus-controlling points; and he realized the handicap of crowded motor-generator sets and transformers closely adjacent to railings or too near aisles for rapid movement.

In order to get the benefit of the reduced labor cost per kilowatt of capacity of modern generating units, it is worth while to make sure that subordinate equipment is sufficiently separated to enable it to be operated and maintained without adding needlessly to the station payroll through the comparative inaccessibility or obstructive features of the apparatus and its arrangement. Ample space in which to use tools specially adapted to the adjustment of complex machinery and fittings is of enough value to justify, usually, at least a moderate increase in building cost. In other words, flexibility and compactness are not always synonymous in first-class designing.

## Unreasonable License Laws

Nobody would deny that it is the duty of the state to see that boilers are safe and that they are safely operated.

If a factory inspector makes you provide fire escapes and put guards around gears and belts, if a plumbing inspector insists that the plumbing, even of your dwelling, shall be sanitarily installed, if the fire and insurance authorities can dictate how much combustible or explosive material you can have around and how you must keep it, surely somebody should pass upon the adequacy of steam boilers, to the consequences of the failure of which not only industrial workers, but occupants of office buildings, customers of department stores, guests of hotels, and even the pedestrian upon the street, are subjected.

Nobody would deny that the state—which is the people—should provide for the safety of the people by the inspection of these boilers and of the man who runs them—if that were all that there is of it.

The man who owns boilers and hires men to run them is afraid that this will not be all.

When he sees laws passed, denying to a man the privilege of working at his vocation because he was not born and brought up in the town where the job is—

When he sees rules adopted, denying to an engineer coming into a community, be he the best engineer in the world, anything except a chance to work in a subordinate capacity, with the lowest grade of license—

When he sees examiners acting in collusion with organized labor, to keep down the supply of engineers by refusing men licenses to run steam plants because they cannot design triple-expansion engines—

When he sees engineers' organizations straining the interpretation of existing laws so as to require every laborer about a steam plant to be a licensed man—

He is afraid that the legislation, however beneficent in its declared purpose, however innocent in its seeming intent, is of sinister design and hostile to his interests.

Most of the opposition to the passage of engineers' license laws is inspired by this fear. Most of the difficulties experienced by the A. S. M. E. committee appointed to formulate standard specifications for the construction of steam boilers and other pressure vessels and for the care of the same in service came because it included in its report a recommended form of license law. As one member put it, "this committee, appointed to get up a blueprint of a horizontal return-tubular boiler, tried to involve the society in objectionable legislation." The only way that the report could be gotten through at all was by leaving out all reference to legislation and all suggestions for laws which would bring about the adoption of the Code by the various states.

As we started in by saying, nobody has any tenable objection to the adoption of boiler inspection and engineers' examination laws, when he is satisfied that they are not excuses to add to the difficulties of the employer and subterfuges to obtain unnatural and unfair advantages for the present holders of jobs in power plants under a specious anxiety for the public safety.

The real effort for such laws is sincere and honest, and those who are making it should most forcibly disavow any ulterior purpose in their agitation and frame their proposed legislation so that it cannot be abused.

It is hoped that the American Society of Mechanical Engineers' Code, above referred to, will come into universal use in this country. Its adoption by the various states, for which powerful interests are striving, opens the way to, if it does not go hand in hand with, the examination and licensing of engineers. Those who are seeking legislation to bring this about should unite their efforts with those who are striving for the boiler-inspection laws and make their demand so reasonable and so sensible that they cannot reasonably be denied.

## Effect of High Steam Pressure on Flywheel Risks

In the days when seventy pounds was so usual a steam pressure that it was taken by the committee on boiler trials of the Centennial Exposition as representing average practice, such an occurrence as a flywheel explosion was almost unheard of. The maximum mean effective pressure which could possibly be gotten was that which would result from carrying seventy pounds full stroke. With an initial pressure of one hundred and fifty pounds, the attainable mean effective pressure in case something goes wrong with the governor is more than twice as great. This means that the velocity generated in a given time would be more than twice as great; but since the centrifugal force increases as the square of the velocity, the stress generated in the flywheel in a given time would be over four times as great. An accident which prompt action, under the less strenuous conditions of twoscore years ago, would have headed off, would be likely to be all over now before the attendant could gather his wits, get into motion, and close the throttle.

# Correspondence

## Rate Discrimination in Massachusetts

In the Feb. 23 issue there appeared an item dealing with the attempt of the New England Power League to get a bill through the Massachusetts Legislature tending to make more equitable the rates for electricity. Perhaps it would be interesting to your readers to learn more as to why the League takes this attitude.

The schedule of the Boston Edison Co. appears to be the only one which is susceptible of having curves plotted which mean anything; but the conditions existing in most of the other electric companies in the state are the same as shown by the curves of the Edison company. There is no difference, so far as results are concerned, between a charge for the privilege of being connected to the lines plus a charge for the power delivered, and charging a high price to the small consumer and a low price to the large consumer. Twelve cents a kilowatt-hour to a small consumer and one cent to the large one represent the range in power costs in Massachusetts. This applies to both demand plus a current charge and a straight charge.

To illustrate the Boston Edison schedule, Fig. 1 has been drawn, which shows that from 0 to 15 kw. there is a demand charge of $60 per kilowatt per year. From 15 kw. upward, this demand charge decreases, and at 10,000

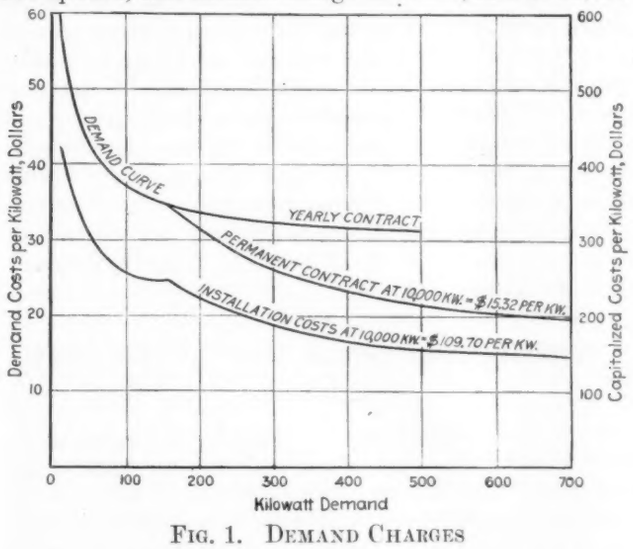



Fig. 1. Demand Charges

kw. demand is $15.32 per year. This applies to permanent rates, that is, to those who make a permanent contract for more than one year. If the contract is only for a year, the reduction is not so great; but even under these conditions, at 500 kw. the demand charge is only $31.20.

Capitalizing these charges, the lower curve is obtained, which would indicate that from 0 to 15 kw., the cost per kilowatt of demand is $428; whereas at 10,000 kw. it is only $109.70. This in itself is rank discrimination, because it is impossible that a city plant could be built with the necessary distributing lines at $109.70 per kilowatt. It is even worse than it appears on the surface, because of the diversity factor.

It has been proved by one of the largest companies that the diversity factor of the large load is practically unity, whereas that of the small load ranges between 3 and 4; that is, a large consumer will usually require in the power

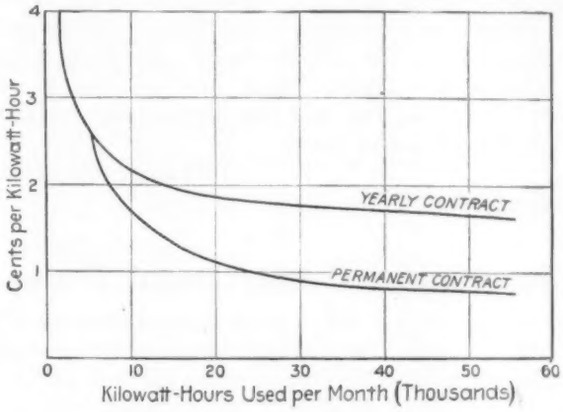



Fig. 2. Prices for Energy Used

plant a capacity equal to his maximum demand, whereas the small consumer will require an installation in the power plant equivalent to only one-third or one-quarter of his maximum demand. As the large consumer is charged for a value less than one-third of the actual cost of the apparatus, the small one on the other hand is charged at a value of three to four times those shown on the capitalized demand costs, or from $1282 to $1712 per kilowatt of actual station capacity.

These figures alone are sufficient to make the New England Power League and others feel that it is time some laws were passed which would make an equitable division of the power costs.

Another item of discrimination is the division between yearly and permanent contracts. It will be noted from Fig. 1 that from 15 to 155 kw., yearly and permanent contracts are the same, but that above 155 kw. there is a decided difference. To my mind this is discrimination, because there should be no point at which these two contracts should be the same if there is any reason for a division between the two characters of contract. If it costs more to handle yearly customers than it does permanent ones, it will cost more for any character of load.

So far attention has been called only to the demand, and it will be noted that there is a wide variation in demand charges. Let us look now at the energy used. Fig. 2 shows the variation in these charges for various consumptions in kilowatt-hours per month. It will be noted that here again there is a marked decrease in cost solely for an increase in the amount of current used. It would seem that if there were a large difference in the cost for the demand, there should be a uniform price for the current, for certainly the demand charges as laid out cover much more than the fixed charges on the small load, and do not cover nearly as much as the fixed charges on the large load. Therefore, in order to even matters up, it would seem as though the power charges for current

delivered should be uniform. They are not, and again the small consumer gets the worst of it. Up to 1500 kw.-hr., he has to pay 5c. per kw.-hr. From this point on, the rate decreases to a marked extent, until at 55,500 kw.-hr. the price is under 2c. for permanent rates. For yearly rates, however, the price is approximately 2.6c. Here again there is a division between the yearly and the permanent rates. As has been stated before, if there is any reason for a division, it should apply to all characters of load; but it does not. The division takes place at 5500 kw.-hr. per month.

In my opinion there is no question as to the injustice and discriminatory feature of the demand charge as outlined, as well as the current charge; and I am heartily in favor of any attempt which may be made to bring about a more uniform and just arrangement of charges for electric light and power.

HENRY D. JACKSON.

Boston, Mass.

## Priming a Centrifugal Pump

This is a problem which confronts many operators of centrifugal pumps. In the Mar. 2 issue there is an article on this subject by J. F. Jones, in which he describes the trouble he had in priming a 30-in. pump. While his method is all right, it does not exhaust all of the air from inside the casing, and no doubt there is considerable "rumbling" in the pump while it is in operation, with a possible chance of injuring the impeller. By referring to Fig. 2 of Mr. Jones' article, it will be seen that it is impossible to remove all the air from inside of that part of the casing which is above the point where the ejector suction enters the suction elbow on the pump. By running with air entrapped inside the casing, the delivery is no doubt reduced.

When a pump such as that shown by Mr. Jones is started with air inside the casing, it is difficult to get rid of all of it through the discharge pipe. What might be a more satisfactory way to prime this pump would be to connect the ejector as shown in Fig. 2 of Mr. Jones' article, except that the suction should be cross-connected to the top of the casing. The method of starting would be the same as that adopted by Mr. Jones, except that after the pump is running and before stopping the ejector, the latter's suction to the top of the pump should be opened and that to the elbow closed. This will exhaust the air from the top of the casing. In this way the delivery should be better than if there were some air in the casing.

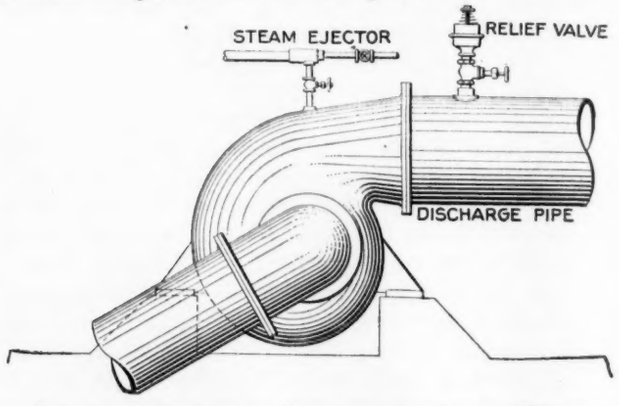



FIG. 1. POSITIONS OF EJECTOR AND RELIEF VALVE

It is not always best to leave off the flap valve, and it is often necessary to have one on the end of the discharge pipe. When a flap valve is used it is necessary when shutting down, to have some way of breaking the vacuum in the discharge pipe which is due to the water running back down the pipe and through the pump.

The possibility of collapsing the discharge pipe on a pump such as that described by Mr. Jones, can be avoided by placing a relief valve near the flap valve on the discharge pipe. The position of this relief valve is shown in Fig. 1. It should be placed on the highest point of the discharge pipe.

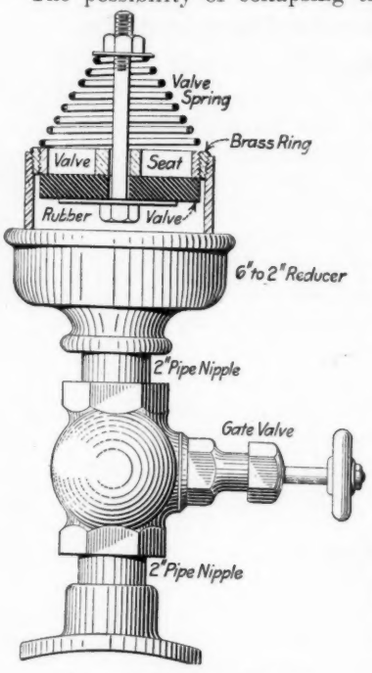



FIG. 2. RELIEF VALVE

A suitable relief valve can be made as shown in Fig. 2. The spring for holding the valve on its seat should be of such strength as to allow the valve to open when a vacuum of about 20 in. of mercury is produced inside the discharge pipe. The opening will permit air to enter, thus breaking the vacuum and preventing the possibility of collapsing the discharge pipe. An ordinary rubber pump valve and seat are shown as making up this relief valve. A gate valve should be placed as shown. This is left open while the pump is running, and if for any reason the pump should stop and the flap valve close, the relief valve would open automatically when the vacuum inside the discharge pipe became great enough to overcome the pressure of the spring.

J. E. POCHE.

New Orleans, La.

## Some Reasons for Different Rates

In the issue of Mar. 16, Mr. Seed has an interesting article showing the reasons for different rates and using the cutting and delivery of ice as an illustration.

In his figures it appears that the cost of labor decreases much too rapidly in the handling of the various quantities of ice; also the investment cost. It is hard to conceive that the investment will decrease from $1.50 a ton to 45c. with an increase of four times the amount handled. It costs more to add a story to an icehouse than just the labor and material. Furthermore, the reduction in prices to the various large consumers is not as great as will probably take place in actual conditions, and nothing like as great as in electric sales, where the ratio of the small to the very large consumers' rate is often 10 to 1, and the small consumers' rate to the average, about 6 to 1. Mr. Seed's greatest variation is 4 to 1, when the largest consumer has no delivery cost at all.

In answering his question, "If Jones found it neces-

sary to give up a part of his business and had his choice, which part would he drop?" it all depends on the figures. If he sold two tons at $8 per ton, and they cost delivered $12.30, there would be a gross profit of $3.70. If he sold the next three tons at $4 a ton and they cost him $13.45, he would have a loss of $1.45. If he sold five tons at $2.50 a ton and they cost him $15.75, he would have a loss of $3.25. If he sold ten tons at $1 and they cost him $11.50, he would have a loss of $1.50. Under these conditions which would he choose? These figures correspond more closely with actual electric prices, as is shown by the following table. One of the large companies sold power as follows:

| | Kilowatt-Hours | Receipts | Cost |
|---|---|---|---|
| Motor service | 25,471,000 | $1,114,624 | $743,000 |
| Street railway | 8,319,000 | 142,987 | 242,200 |
| Street lights | 17,739,000 | 781,792 | 516,500 |
| Commercial lighting, small | 24,568,000 | 2,457,000 | 715,000 |
| Commercial lighting, large | 42,495,000 | 1,792,000 | 1,240,000 |
| Other companies | 4,248,000 | 121,530 | 123,700 |

These costs, like those shown in Mr. Seed's illustration, are the average operating costs of the plant and do not take into account interest or depreciation. If the company had to choose which class of customer it would drop, which would seem the more probable—the small one, paying 10c., or the large one, paying considerably less?

I realize that the figures given here do not recognize many differences in expense to the various characters of load, the cost being the average as shown by the Public Service Commission's report. At the same time, I believe that this illustration shows the absolute futility of trying to compare the manufacture and sale of electricity with ice.

Electricity must be sold as it is manufactured, because it is not possible to store it commercially. This being the case, if we have a customer or group of customers who will demand a certain definite load all day, giving unity power factor, the investment is used to its maximum and therefore the price of the power should be low. If, however, another customer comes along who requires power for only a short period, an additional investment is compelled; and as the power is used but a short time, the investment is used but a short time and the fixed charges are high; and also, as the machinery is used but a short time, the labor charges are large, which means increased cost. This latter is to all intents and purposes the condition which exists with the large power users. During the summer they can be run very largely from the plant which is installed to handle the lighting load; but during the winter they cannot, and apparatus which is idle during the rest of the year has to be put into the plant to meet the power load. This being the case, the investment charges against these power users are very large. This will counterbalance any inaccuracy of the figures given above when taken in the sense of neglecting the interest and depreciation charges; they are comparable and accurate if these charges are taken into account.

HENRY D. JACKSON.

Boston, Mass.

---

That all do not reach the same conclusion from the same starting point is well illustrated in the letter of Mr. Seed on page 383. Using the same figures given, if Jones supplied only one of the classes of customers and could take his pick it would be as follows:

To consumers taking 50 to 100 lb.:

| | | |
|---|---|---|
| Two tons at $8 per ton | | $16.00 |
| Cost of two tons at $2.50 | $5.00 | |
| Cost of team | 5.00 | |
| Cost of two men | 5.00 | |
| | | 15.00 |
| Gross profit | | $1.00 |
| Profit per ton, $0.50. | | |

To markets, etc., 200 to 500 lb. per delivery:

| | | |
|---|---|---|
| Three tons at $6 per ton | | $18.00 |
| Cost of three tons at $2.50 | $7.50 | |
| Cost of team | 5.00 | |
| Cost of men | 5.00 | |
| | | 17.50 |
| Gross profit | | $0.50 |
| Profit per ton, $0.16⅔. | | |

For ice-cream factories, etc., 1000 to 2000 lb.:

| | | |
|---|---|---|
| Five tons at $4 per ton | | $20.00 |
| Cost of five tons at $2.50 | $12.50 | |
| Cost of team | 5.00 | |
| Cost of two men | 5.00 | |
| | | 22.50 |
| Gross loss | | $2.50 |
| Loss per ton, $0.50. | | |

To Smith & Brown, 10 tons at icehouse:

| | | |
|---|---|---|
| Ten tons at $2 per ton | | $20.00 |
| Cost of ten tons at $2.50 | | 25.00 |
| Gross loss | | $5.00 |
| Loss per ton, $0.50. | | |

It is seen that the small consumer is the only one who pays much profit alone. Whom would Jones serve if he could get only one class?

Using the figures exactly as Mr. Seed gives them and supplying all classes, the profit per ton is

| | | | |
|---|---|---|---|
| Small consumer | 2 tons, profit | $3.70, per ton | $1.85 |
| Markets, etc. | 3 tons, profit | 4.55, per ton | 1.51⅔ |
| Ice-cream factories | 5 tons, profit | 4.25, per ton | 0.85 |
| Smith & Brown | 10 tons, profit | 8.50, per ton | 0.85 |

Jones will sell more ice to small consumers than to the other classes, and if he could only sell 1000 tons it is easily seen to which class he would prefer to sell.

I still fail to see why the large consumer should get all the benefit of cheaper production or pay so much less profit per unit.

HARRY D. EVERETT.

Washington, D. C.

## Testing Small Centrifugal Pumps

In the Mar. 16 issue M. R. Blish makes two mistakes in defining the method of computing the head on a centrifugal pump. On page 372 he states that the total head is the sum of "the suction head, pressure head, and the vertical distance between the center of the pressure gage and the point of attachment of the mercury gage." The first error is the omission of a velocity-head correction. It is quite common for the suction pipe on a centrifugal pump to be a size larger than the discharge pipe, in which event we should have to include the difference between the velocity heads in the suction and discharge piping at the points where the gages are attached. This may be proved in various ways, but may readily be seen if we but realize that it requires power to increase the kinetic energy of the water as well as to increase its pressure. Only where the two pipes are of the same diameter, so that the velocity-head correction is zero, would Mr. Blish's definition apply.

The second error is that here, as well as on page 371, he implies that the indication of the mercury manometer is the value of the pressure at the point of its attachment to the suction gage. It is well known that a pressure gage reads the pressure found within itself and that

its height must be considered in finding the value of the pressure at the point of attachment. The same is true of a suction gage. If the suction head is read by means of a vacuum gage, the definition of total head should be corrected to read, *and the vertical distance between the centers of the pressure and vacuum gages.* If the suction head is read by a mercury manometer, the point corresponding to the center of the vacuum gage will be the top of the mercury column on the side of the U-tube that is connected to the pump.

The words in italics are based upon the assumption that there is a continuous column of water between the vacuum gage or the mercury manometer and the point of attachment to the suction pipe. Such a condition may be realized if the connecting tubing be filled with water before the pump is started and if the point of attachment is not at a place where a pocket of air may accumulate. If the connecting tubing contains air only, then Mr. Blish's method of computation would be correct. This is because the weight of the column of air between the mercury manometer and the suction pipe is negligible and hence the manometer reading would be the value of the pressure at the point of attachment. In order to maintain this condition, however, it is necessary to have some means of permitting air to be drawn into the connecting tubing during a test, allowing water that has accumulated to be drawn back into the suction pipe. The figure shown in the article does not indicate any such provision. Unless one does have some means of admitting a small quantity of air during the test, it is better to fill the connecting tubing with water. Otherwise, if the tubing contains water and air both, it will be difficult to properly compute the true suction pressure. In the case of a pump delivering water under a high head and where the vertical distance from the suction gage to the suction pipe is small, the error introduced may be negligible, but under other circumstances it might be appreciable.

Objection must also be raised to the arrangement of suction piping shown in Fig. 3, on page 371. It is a fundamental principle that a suction pipe must contain no summits, otherwise air will accumulate and finally interfere with the flow of water or cause it to cease.

Mr. Blish seems to have the impression that the insertion of baffles in a weir box is to "prevent a serious velocity of approach." Baffles are used to quiet the water and cause it to flow uniformly, but have no effect upon the velocity of approach. The latter is determined solely by the dimensions of the cross-section of the box.

In connection with measuring the rate of discharge of the pump, the author might have mentioned the most common method used in testing, which is by means of a calibrated nozzle on the end of the pipe. Also the venturi meter is a valuable device for such purposes.

In starting a centrifugal pump, the author states that after priming, "open the throttle valve in the delivery pipe and start the motor." While this may be permissible with proper starting devices at the motor, a better procedure would be to bring the pump up to speed before opening the discharge valve. In that way a smaller load would be thrown on the motor at starting, since the horsepower with the valve closed is about one-third that required with the valve wide open.

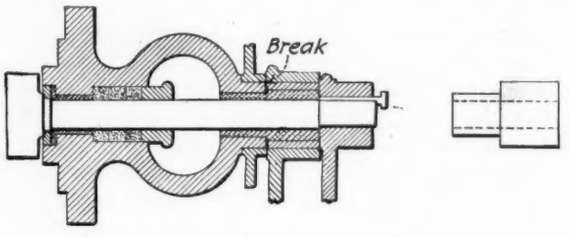
The efficiency curves shown in Fig. 7 for speeds of 1200, 1400 and 1650 r.p.m. are consistent with one another, but the curve for a speed of 1700 r.p.m. is somewhat doubtful. When there is an increase in the maximum efficiency of only about 3 per cent. for a range of speed from 1200 to 1650, it is hardly likely that there will be a drop of about 8 per cent. in passing from 1650 to 1700 r.p.m. Further, it will be noted that the maximum efficiency at each speed is attained at greater values of the rate of discharge as the speed increases for the first three speeds, but for 1700 r.p.m. the maximum efficiency is shown as occurring at a smaller rate of discharge than at 1650 r.p.m. This is hardly reasonable and makes one suspicious of the accuracy of the test data used.

R. L. DAUGHERTY.

Ithaca, N. Y.

## Repairing Corliss Valve Bonnet

One of the admission-valve stems and bonnet bearings on a Corliss engine got overheated from running dry, until it gripped and broke the bonnet, as shown, A good repair job was done by boring out the bonnet and making a




VALVE BONNET WITH BUSHING

sleeve, or bushing, a driving fit. After forcing it into the bonnet it was turned down and bored out to fit the valve stem.

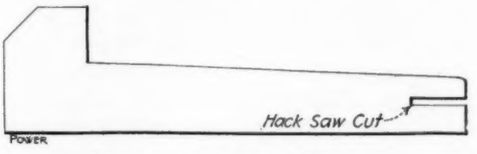
When the parts were reassembled very little adjustment was needed to set the valve to the builder's marks. The job was done in four hours by two men.

JOHN POWERS.

New Bedford, Mass.

## Putting Liners in with Keys

When a key is too loose and it is necessary to put a liner in with it, there is sometimes difficulty in getting the liner to go in with the key, or if the liner is put in first, to get it to stay in place while the key is being driven. To overcome the foregoing, I saw the end of the key, as shown, and insert the end of the liner, doubled




SLOT IN KEY FOR LINER

if necessary, and bend it back along one face of the key. Whether the liner should be placed on the one face or the other depends on the depth and relative smoothness of the keyways.

CHARLES HERMAN.

New York City.

## Replacing Broken Capscrews

Referring to the subject of the letter by A. G. Solomon, page 414, Mar. 23, this difficulty seems to be quite common. Three years ago I had some trouble with a small ammonia-compressor cylinder held in place by eight 7/8-in. capscrews. The screws would become loose and the cylinder would slip on the bedplate at each stroke.

I overcame the trouble in a way similar to that described by Mr. Solomon, but more simple. I drilled a 3/4-in. hole 2 in. deep at the joint of the cylinder flange and the frame, then fitted a dowel pin and drove it in tight. Since then there has been no difficulty in keeping the cylinder rigid.

L. M. JOHNSON.

Emsworth, Penn.

---

In the issue of Mar. 23, page 414, I notice an account of a method used to prevent the breaking of the capscrews used to secure the cylinder of an ammonia compressor to the base.

No doubt, putting in tight-fitting keys would prevent breaking more bolts, but if this means were used on a steam cylinder, I am afraid that before a great while the base would be broken, unless some provision were made for the expansion and contraction of the cylinder, which apparently was not done in the case of the ammonia cylinder.

To fit bolts snugly into the holes is wrong, in my opinion, for the cylinder is heated and expands, while the base remains practically cool and expands very little, thereby putting a shearing stress on the bolts. If the bolt holes had been elongated about 1/8 in., I believe the trouble would have been overcome without the expense of putting in the keys. I would advise using only one key in any case, and enlarging the holes in the other end to allow for expansion, as stated.

H. S. MELLEN.

Philadelphia, Penn.

## Gas Explosions in Boiler Furnaces

I would like to bring to your attention a recent accident in one of our boiler rooms, with the suggestion that it be published, and commented on by readers.

The boilers were of the vertical water-tube type, provided with two sets of fire-doors with automatic stokers (underfeed) between them. Forced draft for all the boilers was furnished by an automatically regulated engine-driven blower, supplemented by natural draft from the stack.

Owing to some minor troubles it had been necessary to cut out one of the two boilers in operation the night before the accident and it, together with the third boiler, had both the natural and the forced draft cut off by the dampers, leaving only one boiler in service to carry the load. This was being forced, with the stack damper wide open, the blower running at full capacity and the stoker rapidly feeding coal. The fireman in cleaning the fire (with only one fire-door open) found a clinker had formed over the tuyeres. As he loosened this an explosion occurred in the firebox, precipitating him backward against the coal bunker and burning him severely. The engineer, who was standing directly in front of the other fire-door of the same boiler, was not injured, as the safety latches on the latter (which had been installed after two similar occurrences of a minor nature) prevented its opening, thus justifying their adoption. But the cleaning-out door at the base of the stack was blown open by the force of the explosion, which appears to have been considerable.

Presumably, this was a carbon-monoxide explosion facilitated by the sudden inflow of air through the tuyeres when the latter were cleared. The lower limit of the explosive range of carbon monoxide in air is in the neighborhood of 15 per cent., and it is difficult to understand how such a rich mixture as this could have remained in the firebox with one fire-door open and the stack draft in operation. We might add that all ashpit doors were sealed and that it is the practice at this works to keep all stack dampers wide open on boilers which have fires under them. When it is necessary to cut out a boiler the forced draft is shut off, and as soon as the fire is dead the damper is closed.

We referred a description of this accident to three prominent companies, one of them supplying forced-draft equipment with turbine blowers, one supplying forced-draft equipment with steam blowers, and the third a well-known boiler insurance company; also calling their attention to a letter in POWER of Mar. 18, 1913, on a carbon-monoxide explosion. Their comments are in part as follows:

First Letter—

We never had any experience of this sort, nor have we ever heard of any with the exception of one case in the writer's early experience with forced draft.

The power plant consisted of six water-tube boilers with which, on account of poor draft, it was difficult at times to keep up steam pressure. The ashpit doors had been removed. The writer recommended forced draft and made a test on one boiler with a turbine blower which was very noisy. To reduce the noise we decided to build a duct around the blower and run it up to within 10 ft. of the boiler-house roof, but as we were not sure that this would give the desired results, we built it of wood and had it lined with shoddy and used cardboard to keep it in place.

During the night the fireman was not able to keep down the steam pressure even with all dampers closed, something which had never happened before, and he closed the ashpit doors of the one boiler which was equipped with the blower, and no doubt also opened the fire-doors of all the boilers. After a while there was an explosion, and the duct caught fire and also set fire to the roof of the boiler room. The writer's explanation of this explosion was as follows: The ashpit doors were closed and the fire-doors open. The damper of the boiler was closed. The duct acted as a chimney and the air went through the fuel bed, caused the formation of carbon monoxide, which went into the ashpit through the blower and up the duct. This heated up the duct, and through leaks around the blower enough air got into the duct to mix with the carbon monoxide and cause the explosion. This experience and other trouble with this particular blower were among the causes that led to the inception of our present type of blower.

We have in the neighborhood of a thousand installations and, as stated before, we have never had trouble on account of carbon monoxide, though many of our installations are operated automatically, so that when the steam pressure goes up both the blower and the damper close and finally shut down completely. Many other forced-draft systems are operated in the same manner.

The letter to "Power" to which you refer is quite interesting, and the only explanation that the writer can give for such an occurrence is that a steam-jet blower was used. This type of blower is small in diameter and the air goes through at a high velocity, and consequently, when shut down little air goes through it by the draft of the chimney. No doubt the damper was also closed, and perhaps just prior to the blower being shut down there was a high rate of combustion and a high temperature of the fuel bed. On account of all this the combustion continued, but not enough air being present, it was incomplete, and carbon monoxide was produced.

With a turbine blower conditions are different. When it is shut down there is enough air going through the fan casing

through natural draft to cause complete combustion provided the damper is not entirely closed. We do not recommend stopping the blower, but we have never thought of the carbon-monoxide danger, but simply do not recommend it on account of believing it desirable to change as little as possible the temperature of the fuel bed. In other words, we recommend that the blowers be run at the slowest rate for the lightest load and be speeded up when the load increases.

We do not think it possible that a high percentage of carbon monoxide can be reached with the dampers open and the fire-doors and ashpit doors closed, and we do not think it possible even with the damper closed provided the blower is not being run at full speed at one time and shut down completely afterward. It should be run at as near as possible the same speed all the time, this speed of course depending upon the variation in demand for steam. We have never considered it unsafe to automatically close off the blower when the boiler pressure reaches a certain point, though, as stated before, we do not recommend it. But in view of the two experiences mentioned in your letter we would not now consider it safe to shut the blower completely and the damper at the same time when the temperature of the fire is high. If the damper is open there will be no danger when the blower is stopped provided the blower is large enough so that the air can pass through it by natural draft.

Second Letter—

We have your favor referring to the explosion in a boiler at one of your works. We have heard of this occurring. The case we have in mind was one in which the damper in the stack or main breeching was closed at the time the explosion took place. We can readily conceive of such a condition, that is, damper closed and gases being given off by the coal with insufficient air supply, with the result that a large volume of CO forms. Then when more air is admitted, although the blower is started, it may be possible for an explosion to occur from the higher temperature resulting with a renewal of combustion, particularly if the stack damper is closed and the gases confined.

Apparently, in your case the damper was open, since you point out that the clean-out door in the base of the stack was forced open by the explosion. This would appear to indicate that your stack was small and that the suction was insufficient.

Such explosions are much more likely to occur in the combustion of soft than of hard coal, on account of the presence of the hydrocarbon gases, which are so much more volatile. We are not inclined to believe that so high a percentage of CO can be formed with the damper open, even though both the ash- and fire-doors be closed.

With regard to operation, the method we recommend is the automatic action of the main blower line by means of a balanced valve and automatic regulator, with a bypass around the former so as to keep the blowers going continuously, the regulator taking care of the slight fluctuations in the load. If the load is extremely variable, then it may be possible to install a balanced valve in the bypass where the throttle valve is, this balanced valve being controlled by another regulator to shut off at a point, say 5 lb. higher than the first regulator. If, on the other hand, the periods of high and of low loads are known and are not too frequent, then this condition can be best corrected by opening, more or less, the throttle valve in the bypass to the balanced valve in the blower line.

Third Letter—

We have had considerable experience on gas explosions, due to carbon monoxide from soft coal, from natural gas, etc., but are unable to advise you definitely as to the lowest range or percentage of carbon monoxide and air that would result in an explosion. While this particular feature is worth investigation as of scientific interest, yet the percentage of such air and gas mixtures in a furnace will, of course, vary widely.

Assuming conditions as described in your letter, we presume the fire-doors, etc., were so arranged that no air was permitted to enter above the fire, the supply being from the blower system. Therefore, with the tuyere openings blocked with clinkers and no air admitted above the fire, carbon monoxide would be generated faster than it could escape, and the furnace, and even the chimney (with one boiler chimney), would be filled with carbon monoxide, which would rapidly be built up to a point where it would be explosive.

It seems if the damper had been open, say 50 per cent., when the fireman opened the fire-door the draft would have quickly carried away the carbon monoxide and thus averted the accident. The matter of closing dampers from 75 to 95 per cent. while the furnace is loaded with coal which is setting free its volatile constituents strikes us as wrong. In our opinion the damper should have a limit closure when a boiler is in service.

In the last comment it is evident that the writer overlooked the points that the opening of the fire-door supplied air above the fire and that if the clean-out door in the base of the stack was blown open, the damper must necessarily have been open—as was actually the case. As for the amount of CO necessary for explosion, Von Schwartz, in "Fire and Explosion Risks," cites Professor Bunte, of Carlsruhe, as authority for 16.6 per cent. and 74.8 per cent. as the lower and upper limits of the amount of CO in admixture with air which will permit explosion when ignited by an electric spark, at the same time stating that the explosion of such can be prevented by the addition of 7½ to 10 per cent. $CO_2$. Von Schwartz also gives the approximate figures of 13 to 75 per cent. for ignition by flame and 636 to 814 deg. C. as the temperature at which pure CO will explode without air.

From a preventive standpoint the comments quoted do not give much information, and we are at a loss to know how to guard against explosions of this character with this particular installation. We have experienced dust explosions from the loosening of deposits of soot in the flues, oil-gas explosions with oil-fired systems, carbon-monoxide explosions from negligence in completely closing stack dampers and also severe accidents from flarebacks from boilers equipped with steam-blower systems—the latter caused by too high steam pressure or by firing the coal too far back in the furnace, where it obstructed the draft. For all these there seem to have been reasonable remedies, but for a case such as that mentioned we see no cure other than instructions to keep the tuyere openings clear of clinkers, which was, after all, what the fireman was trying to do.

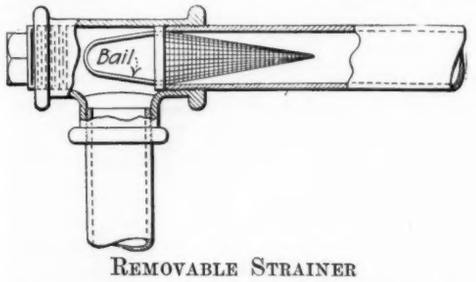
A general discussion by qualified persons would be appreciated.

L. A. De Blois,
E. I. du Pont de Nemours Powder Co.
Wilmington, Del.

## Strainer in Pump Suction

I have found that a strainer, made as shown in the illustration saves a lot of trouble by preventing small sticks and stones entering the pump and lodging under the valves.

The strainer is made fast to a brass ring, which will hold against the end of the pipe and keep the screen from




Removable Strainer

being drawn into the pump. A handle, or bail, is convenient to pull the strainer out for cleaning. A tee is used in the suction pipe instead of an elbow, and the strainer put in as shown, and a plug is used to close the end of the tee and to make the appliance easily accessible. The mesh of the screen should be suitable to the size of the pump and the kind of service. Brass or copper wire is preferable, but galvanized wire will answer in some cases.

Ithaca, N. Y. John P. Kolar.

# Inquiries of General Interest

**Side for Connecting Equalizing Switch**—On which side of an electric generator should the equalizing switch be connected?

W. H. M.

An equalizing switch is to be connected on whichever side of the generator the series field winding is placed.

---

**Vacuum Not Ascertainable from Temperature**—Would determination of temperature of the condensate discharged from a condenser be an accurate method of measuring condenser pressure by ascertaining pressures corresponding to temperatures given in the steam tables?

A. J. F.

The temperature of the condensate would not be a true indication of the pressure, on account of the presence of air in the condenser.

---

**Backfiring Trouble**—Would placing a wire screen in the intake pipe of a gasoline engine between the carburetor and cylinder prevent backfiring?

C. E. S.

A screen would undoubtedly prevent backfiring through the intake pipe, but would not remove the cause of imperfect operation of the engine. The backfiring may be due to wrong timing of the ignition, improper seating of the inlet valve, thus allowing the exploding mixture to get by it into the intake pipe, or a combination of a lean mixture and too early ignition.

---

**Measure of Ductility**—What is the measure of the ductility of a metal?

E. H.

The percentage elongation and the percentage reduction of area are usually considered together as the measure of the ductility of a metal. After a bar of the material under tensile stress has passed its elastic limit it begins to be permanently elongated in the direction of the pull. The increase in length multiplied by 100 and divided by the original length is the "percentage elongation." When a bar is elongated it shrinks in cross-section, and just before it breaks it usually "necks down" at the point of fracture. The original cross-sectional area minus the area of smallest cross-section after fracture is called the "reduction area," and this difference multiplied by 100 and divided by the original area is the "percentage reduction of area."

---

**Increase of Bolt Tension after Cooling**—If a 1½-in. diameter steel bolt is drawn up tight when at the temperature of 160 deg. F., how much will its tension be increased after it has cooled to 70 deg. F.?

R. B. N.

If there is no yielding of the head, nut or screw from compressive stresses, the additional tensile stress in the bolt due to cooling will be the same as that required for elongating the bolt as much as it would elongate or contract for the same change of temperature. The coefficient of lineal expansion or contraction of steel is 0.0000065 of its length per degree change of temperature, and therefore the elongation or contraction per inch of length for a change of 160 — 70 = 90 deg. F. would be

$$0.0000065 \times 90 = 0.000585 \text{ in.}$$

As for steel, the modulus of elasticity, or load per square inch of cross-sectional area divided by the extension per inch of length, is 30,000,000, then for an extension of 0.000585 in. the stress would be

$$0.000585 \times 30{,}000{,}000 = 17{,}550 \text{ lb.}$$

per square inch of cross-sectional area, and for 1½-in. bolt, or a cross-sectional area of

$$1\tfrac{1}{2} \times 1\tfrac{1}{2} \times 0.7854 = 1.767 \text{ sq.in.,}$$

the stress would be

$$17{,}550 \times 1.767 = 31{,}010.85 \text{ lb.}$$

---

**Size of Feed Pump**—What size of single feed pump should be employed for a boiler which requires 3850 lb. of feed water per hour?

W. H. M.

A uniform feed-water supply would be equivalent to

$$3850 \div 8\tfrac{1}{3} = 462 \text{ gal.}$$

per hour, requiring a pump displacement of

$$(462 \times 231) \div 60 = 1778.7 \text{ cu.in. per min.}$$

But to meet emergencies the rated capacity should be about double the delivery required for uniform rate of feeding, i.e., neglecting slippage and reduction of piston area due to the piston rod, the displacement capacity should be about 3558 cu.in. per min. Allowing a maximum speed of 60 strokes per minute the piston displacement should be $\frac{3558}{60}$, or about 60 cu.in. per stroke.

To determine the size of water cylinder, either the area of the piston or the length of the stroke must be selected. Neglecting slippage and reduction of the area of piston due to the piston rod and assuming the diameter of water cylinder as 3¼ in., the area of the piston would be

$$3\tfrac{1}{4} \times 3\tfrac{1}{4} \times 0.7854 = 8.2958 \text{ sq.in.}$$

requiring a stroke of

$$\frac{60}{8.2958} = 7.23 \text{ in.}$$

and for most practical purposes the commercial size, 5½x3¼x7 in. (5½-in. diameter of steam cylinder, 3¼-in. diameter of water cylinder, 7-in. stroke) would answer.

---

**Pressure Required for Running Noncondensing**—If an engine, operated with steam at an initial pressure of 120 lb. absolute, ¼ cutoff and 26 in. vacuum, loses its vacuum, what initial pressure would be required to operate the engine noncondensing with the same length of cutoff, the valve setting and load remaining unchanged?

S. S.

The mean forward pressure per pound of initial is given approximately by the formula

$$P_m = (1 + \log_e R)\ (f + c) - c$$

in which

$P_m$ = Mean forward pressure per pound of initial (absolute);

$\log_e R$ = Hyperbolic logarithm of the ratio of expansion;

$R$ = Ratio of expansion $= \dfrac{1+c}{f+c}$

$f$ = Fraction of stroke completed at cutoff;

$c$ = Clearance per cent. of piston displacement.

Assuming 5 per cent. clearance, the ratio of expansion would be

$$\frac{1+c}{f+c} = \frac{1+0.05}{0.25+0.05} = 3.5$$

and as the hyperbolic logarithm of 3.5 is 1.2528, the mean forward pressure per pound of initial would be

$$(1 + 1.2528)\ (0.25 + 0.05) - 0.05 = 0.6258 \text{ lb.}$$

and for 120 lb. initial absolute the mean forward pressure would be

$$120 \times 0.6258 = 75.1 \text{ lb.}$$

When running condensing with 26-in. vacuum the back pressure would be

$$30 - 26 = 4 \text{ in. mercury,}$$

or

$$4 \times 0.491 = 1.964 \text{ lb. absolute,}$$

and the m.e.p. of an ideal diagram would be

$$75.1 - 1.964 = 73.136 \text{ lb. m.e.p.}$$

but when running noncondensing the back pressure would be about 1.5 lb. gage, or about

$$15 + 1.5 = 16.5 \text{ lb. absolute}$$

and to obtain the same m.e.p. (73.136 lb.) for the ideal diagram the mean forward pressure would need to be

$$73.136 + 16.5 = 89.636 \text{ lb.}$$

If in each instance the actual is the same per cent. of the ideal diagram, and as the mean forward pressure when cutting off at ¼ stroke is 0.6258 lb. per pound initial absolute, then to realize a mean forward pressure of 89.636 lb. absolute would require

$$89.636 \div 0.6258 = 143.2 \text{ lb. absolute,}$$

or about

$$143.2 - 15 = 128.2 \text{ gage pressure.}$$

---

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—EDITOR.]

# Convertible Combustion Engines*

By Alan E. L. Chorlton

One might at first sight say that the combustion engine most ready to work on different fuels would be of the self-ignition type, in which the heat of compression is sufficient to ignite the incoming fuel, and the only change necessary in going from liquid to solid fuel would be in the fuel-injection device. In practice, however, owing to the difficulty with solid-fuel injection, such a type would not prove workable, and even when the fuel is first gasified the results do not justify the complication. The problem of designing an engine is better met by trying to combine known types for gas and oil, in which good results are obtained at present and which in general principles show the same characteristics.

In the normal engines for both gas and oil the chief difference lies in the degree of compression. Thus, the compression of a modern gas engine may vary from 90 lb. when using coke-oven gas, to 150 lb. for producer gas; while for the liquid-fuel engine using crude or residual oils the compression pressure may exceed 500 lb., but may be considerably less if the temperature of ignition is obtained by uncooled surfaces or auxiliary or pocket firing is used.

As there is no fundamental difference in engines for gaseous or liquid fuels except in their conventional cycles, it follows that any schemes of convertibility must provide means whereby the requisite compressions can be readily obtained, but as there is a great gap between 500 and 150 lb., the tendency is to combine the lower-compression oil engine and the higher-compression gas engine and thus deal with a smaller compression-pressure range. The desirable characteristics of the convertible engine are: Simplicity and reliability, high economy for each fuel, first cost (little above that of the standard engine), easy convertibility, and as nearly as possible the same power developed for each fuel.

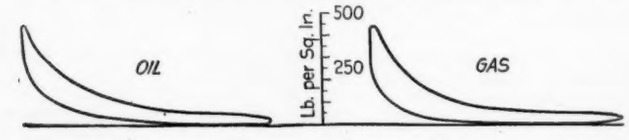



Fig. 2. Indicator Diagrams with Oil and with Gas

Consideration of the subject may be more clearly undertaken by dividing the types of engines in use into three groups, with further subdivisions, due to peculiarities of design: (1) Engines of low compression and low power; (2) engines of high compression and higher power; (3) engines of medium compression and higher power.

GROUP 1—As an example of the first group, take the ordinary motor-car engine having a compression up to 90 lb., and which, with slight modifications, will run on gasoline, good kerosene with an exhaust-heated carburetor, town gas and producer gas. It is designed for and works best on gasoline; fairly well, however, but not so economically, on town gas; requires very good kerosene and is not efficient at the low-compression with producer gas.

To extend the range to use a poorer grade of kerosene or a good grade of crude oil, means must be provided whereby more heat is available for the ignition. This can be conveniently done by the addition of an unjacketed portion to the cylinder head; then the engine becomes of the hot-bulb type. A change of parts for such an engine does not present any difficulties. The gas fittings are provided with electric ignition, which is also suitable when gasoline is used, while the kerosene and good crude oil would be self-ignited by the hot bulb. Fig. 1 illustrates the practical application of these modifications. The results obtained with this engine, are:

| | Compression, Lb. per Sq.In. | Maximum Pressure, Lb. | M.E.P. Lb. | B.t.u. per B.hp.-hr. |
|---|---|---|---|---|
| Kerosene ..... | 55 | 210 | 58 | 14,500 |
| Producer gas ..... | 90 | 230 | 65 | 12,000 |

This type of engine is suitable only for comparatively low powers, and the range of fuels does not include any heavier

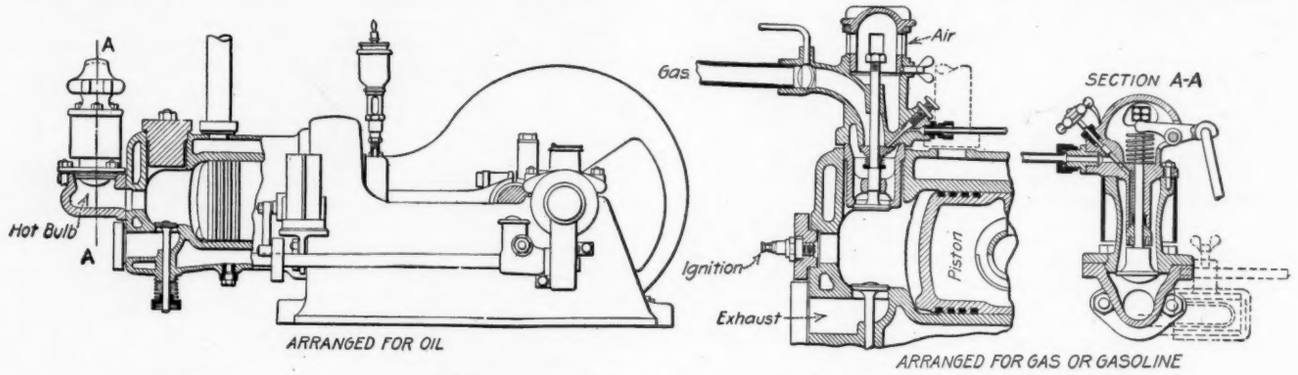



Fig. 1. Convertible Engine for Small Powers Using Gas and Liquid Fuel

than good crude oil. Furthermore, its economy on oil is not high. It is described here because it contains the basic principles of a more suitable engine for large powers.

GROUP 2—This group is represented by the Diesel engine, with a compression of over 500 lb. when using tar oils (unless an ignition oil is used). As the maximum compression for gas normally does not exceed 150 lb., there are mechanical difficulties in building an engine in which both of these com-

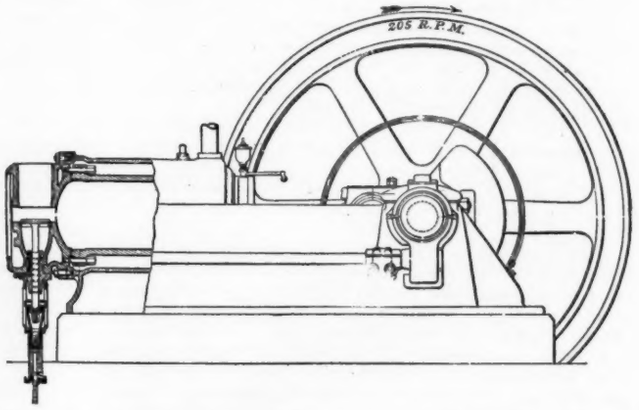



Fig. 3. Convertible Engine with Exhaust Valve Attached but without Head

pressions can be obtained with reasonable modification. Furthermore, the Diesel has an expensive high-pressure compressor which is unnecessary for the ordinary type of gas engine. Apart from mechanical difficulties, we may compare the commercial possibilities of this type, the oil and gas sides being as follows for engines of the same cylinder dimensions:

| | Compression, Lb. per Sq.In. | Abnormal Maximum Pressure, Lb. | M.E.P. Lb. | Approx. B.t.u. per B.hp.-hr. |
|---|---|---|---|---|
| Oil engine.... | 500 | Over 1000 | 100-110 | 8000 |
| Gas engine... | 150 | 600 | 80 | 8500 |

These figures show how incompatible the two designs are, for they illustrate that the Diesel structure must be built twice as strong for the very high maximum possible pressure, owing to the fuel valve sticking, etc. The higher

*Read before the Institution of Mechanical Engineers, London, on Feb. 19, 1915.

mean effective pressure used is some compensation for this extra cost and weight. The mean cylinder pressures reveal a still further disadvantage when the gas conversion is considered; for besides this high first cost, there is a reduced power, owing to working at a lower mean effective pressure—80 as against 100-110 lb. A convertible engine on these lines does not seem a commercial possibility.

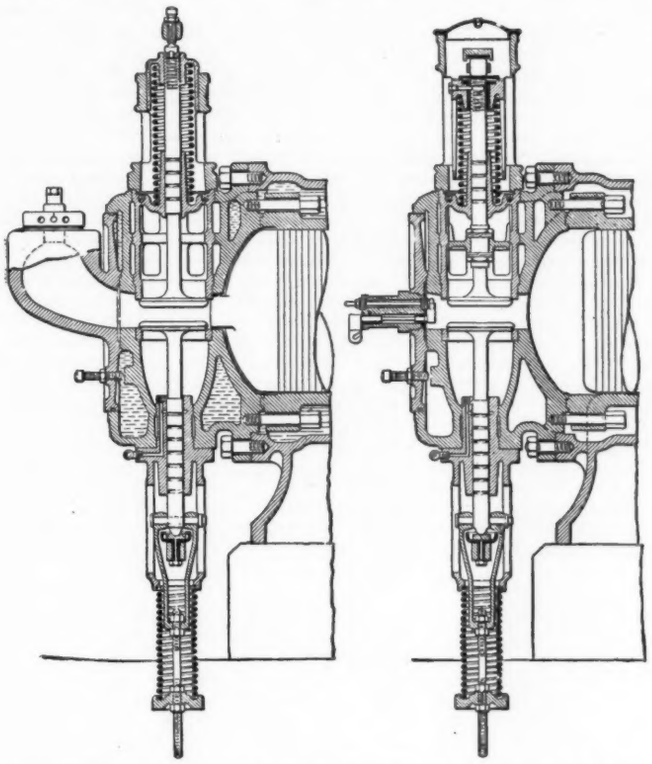


FIG. 4. SAME ENGINE AS IN FIG. 3 ARRANGED FOR OIL AND FOR GAS

GROUP 3—The range of the compression of these engines lies between that of Groups 1 and 2, and usually is from 150 to 300 lb. A compression of 150 lb. is suitable for most forms of producer gas, but because of preignition it is not usually exceeded. On the other hand, it is necessary for all compressions below self-ignition pressure to employ some auxiliary means to obtain the necessary temperature for the proper combustion of the residue oil. Normally, the additional heat is obtained by an unjacketed surface or ignition bulb at the cylinder end.

The various devices concerning temperatures, compression and convertibility of this group may be considered by subdividing it under the following heads:

a. Engines working with a compression not exceeding 150 lb. both for oil and gas, and which may employ pocket-firing with or without air injection for one and electrical ignition for the other, or some combination, the smaller compression not involving material mechanical changes of the parts.

b. Engines working with a higher compression for oil than gas, involving some modification of the combustion chamber by substitution of a part for oil as against a part for gas; otherwise maintaining the simplicity of both types.

c. Engines obtaining the necessary change from gas to oil by temperature control of the air charge, together with alteration of the valve settings.

d. Engines employing the super-compression of Dr. Dugald Clerk, to control effectively the compression required for either fuel. (Final ignition temperature by other means.)

Under subdivision **a** there are a large number of engines used only for oil, but which might, without much alteration, become effective in the use of gas, although the whole combination is not as efficient as the highest individual member.

In considering division **b**, a useful comparison of its possibilities is given in tabular form, for engines of the same cylinder dimensions:

| | Compression, Lb. per Sq.In. | Abnormal Maximum Pressure, Lb. | M.E.P. Lb. | B.t.u. per B.hp.-hr. |
|---|---|---|---|---|
| Oil engine..... | 250-280 | 600 | 80 | 8100 |
| Gas engine..... | 150 | 600 | 80 | 8500 |

It will be seen from this that the outputs and general figures relative to the engine bear a great similarity. Moreover, the diagrams of Fig. 2, taken when running on oil and on gas, are similar. Therefore, the possibilities of this type of convertible engine appear great, and an engine which has been made in considerable numbers to fulfill these conditions is shown in Figs. 3 and 4. (Built by Messrs. Ruston, Proctor & Co., Ltd.) These views show clearly that in both cases the engine presents the ordinary features of the four-stroke-cycle type, and the only change involved in converting from oil to gas lies around the combustion bulb of the oil engine, and in the change from an oil-type to a gas-type piston.

The arrangements in division **c** may be justified for mechanical and constructional convenience, but they can hardly be defended on the score of efficiency. In one method the jacket of the cylinder cover is formed to withstand a pressure, and is worked in the manner of a boiler. The increased heat is impressed in the charge of air during the compression stroke to raise the temperature sufficiently for ignition. This arrangement thus replaces the hot-bulb or unjacketed end of the cylinder. It has some advantages, in that it is perhaps more controllable and the steam generated may be used for some useful purpose, perhaps in conjunction with an exhaust-heated boiler for an auxiliary steam cylinder on the main engine. The water injection of some hot-bulb engines is also done away with. This type of engine is unusual in practice.

When in use as a gas engine, such a convertible machine would have to dispense with the pressure-jacket temperature when using the lower compression and temperature needed for such an engine. Actual heating of the inlet air may be a practical convenience for dealing with a particularly refractory oil not amenable to the available compression of the engine. The valve setting may be modified to give suitable compressions for oil and gas; this can be worked in conjunction with the temperature arrangement just described. The gas engine, however, suffers in loss of output.

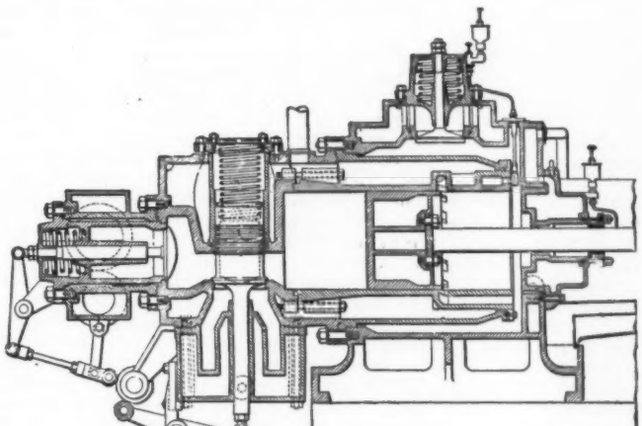


FIG. 5. SECTION THROUGH CLERK SUPER-COMPRESSION ENGINE

The Clerk super-compression engine (Fig. 5) is a much more suitable and promising type for dealing with the variable compression problem of the convertible engine. In this, an extra charge of air or inert gas is added to the working mixture at the end of the suction stroke, by which means much lower maximum flame temperatures are obtained and a higher mean cylinder pressure is rendered possible. As a convertible engine, it is particularly suitable, for by varying this amount of added air the compression can be adjusted between wide limits, the final temperature being controlled by any of the previously indicated means. For instance, with a compression of 300 lb. maximum, the full displacement of the air pump may be used; for the lower compression of the gas engine one can, by any suitable valvular means such as an ordinary bypass, reduce the amount of discharge as required.

CONCLUSION—The conclusions of the author are that, for powers up to say 1000 b.hp., a type such as the Ruston is the most suitable as a convertible engine, while for still larger powers, when tandem engines and size and weight of removable parts become a problem, the Clerk super-compression type offers very interesting and hopeful possibilities in this field.

# Troubles Encountered with Carbon Brushes*

By E. H. Martindale

*SYNOPSIS—The paper is divided into five sections, based on the location of the cause of the trouble: (1) field, (2) armature, (3) commutator, including brush rigging, (4) external electrical, and (5) external mechanical.*

The characteristics of carbon brushes which commonly affect the operation are resistance, hardness, abrasiveness, coefficient of friction, contact voltage drop and heat conductivity. None of these terms needs explanation, but the writer wishes to emphasize the importance of not confusing hardness with abrasiveness. By abrasiveness is meant the scouring or cutting action of the brush. Relative hardness may be judged by cutting the brush with a pocket knife or by marking with it on paper, or if more accuracy is desired, a set of pencils from 2B to 8H will be an aid, as a pencil softer than the brush will mark it and one harder will scratch it. The hardest brush with which the writer is familiar has no abrasive action, while one of the softest has a decidedly abrasive action.

## FIELD TROUBLES

As the field coils are connected in series or series parallel, a partial short-circuit may occur in one coil without materially affecting the heating of the coil. This is usually attended with severe sparking at one or two studs, although in a wave-wound machine the commutation may be little affected. This trouble can best be located by noting the voltage drop across each coil with a constant current through the coils. Similar trouble may be caused by an error in rewinding a field coil. On some machines the shunt fields are connected with two or three fields in series and two or more of these groups in parallel. A partial short-circuit in one coil will then affect the entire group, electrically unbalance the machine, and cause heavy short-circuit currents.

In a cumulative compound machine one series field may be reversed accidentally and, as the load increases, sparking will occur usually at two adjacent studs; or the entire series field by mistake may be connected to oppose the shunt field, which will result in blackening of the commutator, with severe sparking at heavy loads. The best way to detect this is to excite separately the shunt and the series fields, being sure that the current flows in the same direction as when the machine is in operation. The polarity of each pole should be tested with a compass and should reverse as the compass is passed from one pole to the next. Furthermore, the polarity of each should be the same when either field is excited. In a generator the voltage will decrease and in a motor the speed will increase as the load increases.

Unequal air gaps are responsible for much commutation trouble. If one pole face is nearer the armature the flux across the gap is greater and a higher voltage will be developed in the coils under that pole. This may result in heavy short-circuit currents between the studs adjacent to that pole and other studs of the same polarity.

## ARMATURE

An open circuit in an armature coil causes the most vicious form of sparking, accompanied by pitting of the mica between the commutator bars connected to this coil and the adjacent ones. The usual method is to connect an incandescent lamp in a testing circuit, and with two pointed terminals make a bar to bar test on the commutator; the open circuit is shown when the lamp does not light. Similar sparking in a lesser degree may be caused by a high-resistance connection between the end of a coil and the commutator riser. This can be detected by passing a current through the armature and noting the voltage drop between adjacent bars, as the voltage will be higher than normal when the poor connection is in the circuit measured. This method may also be used to detect an open circuit, but a voltmeter reading as high as the voltage impressed on the armature must be used.

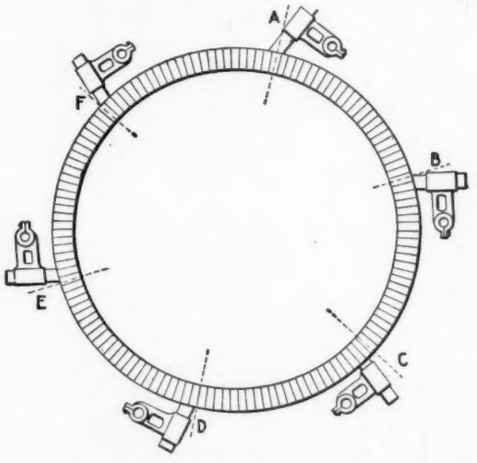
A short-circuit between two sections of a coil, two coils in the same slot, the end connections of two coils, or between the commutator bars, will be evidenced by excessive heating of the coils affected, and unless repaired will sooner or later result in a burned-out coil. The same method may be used as in the preceding case, and the voltage between adjacent bars when the coil is in the circuit will be below normal.

The demagnetizing and cross-magnetizing actions of an armature have serious effects on the commutation of many machines. To get good commutation in a non-interpole machine, the brushes usually must be set ahead of the mechanical neutral on a generator and back of the mechanical neutral on a motor.

If the brushes are shifted far to obtain good commutation, and if the magnetization of the polepieces is not well above the knee of the saturation curve, the demagnetizing effect of the armature may seriously reduce the voltage of a generator or increase the speed of a motor. On the other hand, the cross-magnetizing effect may be sufficient, if the brushes are not shifted to the electrical neutral, to place the coils undergoing commutation in a heavy field, with resultant heavy short-circuit currents, severe sparking at the brushes and all the attendant evils. The remedy is to widen the neutral field by filing away the edges of the polepieces.

## COMMUTATOR

Commutator-brush troubles are numerous, and often difficult to identify. One of the most troublesome problems with non-interpole generators or motors which do not operate at constant load is the difficulty of finding a point at which the brushes will operate at all loads without injurious sparking. This is due to the cross-magnetizing action described.




Unequal Spacing Due to Brush-Holders Being Rotated Too Far

Spring tension, or the pressure with which the brushes bear on the commutator, seldom receives proper attention. The most economical pressure is the lowest consistent with a low contact loss, a clean commutator and freedom from sparking, glowing or pitting of the brushes. It is seldom advisable to use a pressure less than 1½ lb. per sq.in. of cross-section. The writer recommends on stationary machines a pressure of from 2 to 4 lb., depending on local conditions and the grade of brush; and from 4 to 8 for crane motors, haulage motors, railway motors and similar machines.

Brush spacing is important, but is usually neglected. In the sketch the studs are equally spaced and the dotted lines show the correct brush position, but owing to the brush holders on arm A being rotated too far in one direction and on arm C too far in the opposite direction, the voltage generated in section ABC is different from that in AFE or CDE, and this will result in short-circuit currents between A, E and C, high enough to neutralize the unequal voltage.

The magnitude of this short-circuit current may be illustrated by a test conducted a few years ago by the writer on a 400-amp., 250-volt, six-pole shunt generator. On one positive stud the brush holders were rotated to place the brush ¾ in. ahead of the correct position; the other positive studs were left unchanged. With the brushes incorrectly

*Excerpts from paper read before the Cleveland Section, A. I. E. E., March 18, 1915.

spaced but operating at the best point, the short-circuit current was not excessive. When the brushes were shifted two bars away from the neutral and with no external load on the machine the short-circuit current rose to 800 amp., or twice the normal full-load current. As the voltage back of this current was low the actual power loss was small, but the heating of the windings and the effect on the commutator and brushes were serious. This is an extreme case, but a smaller difference in spacing may often be serious.

From the time a commutator bar touches one edge of a brush until it leaves the opposite edge, the current in the coil undergoing commutation should fall from full load to zero, and rise in the opposite direction to full load. If the current in the coil is more or less than that value, the final adjustment comes as a sudden rush of current as the bar leaves the brush. In many machines the time of commutation is less than 0.001 sec. The brushes should therefore be shifted to a point where the coils undergoing commutation are in a field strong enough to make this change. If the brushes are too thin, sufficient time is not allowed for commutation, and if too thick, the coil may over-commutate.

If there is not sufficient clearance between a brush and its holder or if foreign matter becomes lodged in the holder, the brush will not move freely and may make poor contact, resulting in blackening of the commutator, sparking, heating and other evils.

On the other hand, if there is too much clearance and the brushes are not equipped with shunts, the current may pass from the brush to the brush holder through a small arc and cause undue wear of the holders. On machines which have been in service a long time trouble frequently arises from worn holders. This is particularly true if the machine runs in both directions, as the brush face changes when the machine is reversed and thereby reduces the time of commutation and increases the current density in the brush faces. Brushes in worn holders are also more inclined to chatter and chip.

Brushes are sometimes ordered longer than standard with a view to securing a long life, but the spring usually gives a side push and causes trouble, which shortens the life of the brushes and perhaps damages the commutator.

Noise of carbon brushes is due to a mechanical vibration called chattering. If due to the friction of the brush on the commutator, the noise may have various pitches. The remedy is a change in spring tension, angle of operation or grade of brush, although relief may be obtained by lubrication of the commutator at intervals. If the noise results from high mica or wide slots in a slotted commutator, the pitch of the sound will correspond to the number of bars passing under the brush in a second. If the noise is caused by slots, it may be necessary to change the spring tension, the angle of operation, or grade of brush, as lubrication is not advisable on a slotted commutator.

Pitting or honeycombing of the brush faces is nearly always caused by short-circuit currents or a very low brush pressure, but occasionally it is due to insufficient current carrying capacity of the brushes. Many cases of pitting may be corrected by reducing the thickness of the brushes, but it is better to look for some other cause first, as already described.

A loose commutator bar may be flush with other bars when the machine is stationary, but may be thrown out slightly when running, owing to centrifugal force. This will lift the brush and will burn one or more bars just ahead of the high bar, depending on the number which the brush spans, and further, will burn some of the bars back of the high bar, depending largely on the speed of the machine and the brush pressure. Its presence can often be detected by the knocking sound of the bar hitting a brush once every revolution.

In repairing commutators and sometimes in manufacturing them, commutator bars of different hardness are used, and one bar may wear faster than another, causing a flat spot or a high bar.

Blackening of a commutator may be caused by sparking, the use of too much lubricant, or by the character of the brush. Blackening will sometimes occur on every alternate bar or every third bar, corresponding to the number of coils per slot, and may often be shifted to another group of bars by shifting the location of the brushes. This seems to be due to a magnetic kick in the coil undergoing commutation when the armature tooth next to the coil suddenly leaves the field. The remedy is to have the neutral field wide enough to permit the tooth to leave the strong field before the commutator bar comes under the brush.

The best practice in commutator slotting consists in undercutting the mica about $^{3}/_{64}$ in. below the surface of the commutator. It is important that great care be exercised to see that all the slots are free from strips or particles of mica flush with the commutator. It is not advisable to use lubricant or artificially lubricated brushes on a slotted commutator, as the lubricant may get into the slots, collect dirt and cause short-circuits between bars. On slow-speed machines, where the peripheral speed is not sufficient to throw out particles of dirt, the commutator slots should be blown out or scraped out at regular intervals. On a slotted commutator a brush with no abrasive action may be used and will result in long life of the commutator and brushes. A non-abrasive brush or a self-lubricating brush does not necessarily mean a soft brush.

Heating of the commutator on a machine may be caused by any form of sparking, short-circuit currents, friction of brushes, high brush pressure, too low brush pressure causing high contact loss, dirty commutator, overloads, too small commutator, resistance of windings, loose connections, eddy currents and hysteresis. As the ultimate capacity of a machine depends on the allowable temperature rise, it is important to prevent heating wherever possible.

### OUTSIDE ELECTRICAL CAUSES

Outside electrical causes of commutation trouble may be briefly stated as overloads, line surges, and cross currents between two or more machines running in parallel. Where the angular speed of a reciprocating engine varies greatly, surges may occur, caused by a slight reduction in speed of motors on the circuit when the voltage is low and a consequent rush of current when the voltage reaches its maximum. It may be impossible to locate surges or cross-currents without the use of an oscillograph.

### OUTSIDE MECHANICAL CAUSES

If an armature is mechanically unbalanced severe vibration may occur, especially if it is run at high speed. This may produce flat spots, unbalanced electrical conditions, loosening of commutator bars and other serious troubles.

If the machine is on unstable foundations similar troubles may be experienced, owing to vibration of the entire machine. In this class may be placed crane motors and similar machines which, however, are usually designed with this factor in view. Poor belt lacing or uneven gears may produce vibration or strain, with the same results.

## Condenser-Tube Corrosion

The third part of the Institute of Metals' "Contributions to the History of Corrosion" has been issued. It consists of a contribution by Arnold Philip, Admiralty chemist, attacking the conclusions of G. D. Bengough and R. M. Jones, and a supplement containing their reply. An abstract of the first two parts appeared in "Power," Dec. 2, 1913, page 781.

We do not reproduce the third part at length, as it seems largely to reinforce Bengough and Jones' original work and conclusions, namely, that entrained pieces of coke are not grave causes of condenser-tube corrosion. Some of their minor points concerning the technique of their experiments may, however, be of interest.

The point is made that the method of determining loss of weight is of little use in investigating corrosion. Of all failures, 90 per cent. "are caused by local dezincification, and when once this has been started loss-of-weight methods are useless. The action was quite local and irregular, and the white zinc salt was strongly adherent and usually could not be removed without injury to the underlying oxide layer."

The authors conclude that, consequently, a means of detecting dezincification is the only rational way of detecting the progress of corrosion. Here they leave us at sea. The only method available giving actual measurement seems to be microscopic investigation, especially of sections cut at right angles to the supposed dezincified spot. This involves destruction of the specimen. The next best is a hardness test by rubbing with a blunt steel needle, the dezincified area being softer than the unchanged brass. In fresh specimens the eye with proper training can detect the dezincified spots, even in the early stages; in old specimens the copper oxidizes, and this copper cannot be told from the ordinary oxide layer.

Another point made is that when various plates of metal are being tested for resistance to corrosion it is not fair to take pieces of tube and flatten them, or clean them, or anneal, or in any other way alter the normal skin of the metal. After attention has been called to this it seems self-evident, yet we have known cases where it has not been observed.

Another point spoken of is that sea water to which sufficient sodium carbonate has been added to make it alkaline in reaction instead of neutral, as is normal sea water, is much more active in producing dezincification than the normal sea water. Here is a practical point in condenser practice.

## St. Louis Consumers Protest against Rates

The Engineers' Incitation Club, representing consumers of electricity in St. Louis, has filed a petition with the public-service commission of Missouri, protesting against the alleged discriminatory rates charged by the Union Electric Light & Power Co., of that city. The present maximum rate to small consumers is 10c. per kw.-hr., whereas it is charged that certain large consumers pay less than 1c. per kw.-hr. The petitioners would have the rate fixed at a maximum of 5c. per kw.-hr. for the first 120 hr. of installed capacity used per month, with all in excess thereof at 2½c. per kw.-hr.

It appears that the Union Electric Light & Power Co. procures a large portion of its supply from the Keokuk hydro-electric development. This power is generated by the Mississippi River Power Co., but instead of purchasing direct from this company the defendant buys through intermediary companies known as the Mississippi River Power Distributing Co. and the Electric Co. of Missouri. It is alleged that these corporations are under common control and through collusive agreements are defeating the purpose of Congress in making the water-power grant, and instead of the public being the beneficiaries of low rates, a few promoters and stockholders in the companies mentioned are reaping the benefit.

## ENGINEERING AFFAIRS

**American Association of Refrigeration Meeting**—The fifth annual meeting of the American Association of Refrigeration will be held at the Hotel Astor, New York City, May 11 and 12. There will be important reports from officers and standing committees and commissions of the association, including a detailed financial statement of the Third International Congress of Refrigeration.

**The Ohio Society of Mechanical, Electrical and Steam Engineers** will hold its next meeting June 17 and 18, at Toledo. Plans are being made for an outing, together with the regular reading and discussion of papers. Among papers to be given are: "Firebricks for Boiler Settings," by W. G. Heisel; "Some Features of the Cleveland Municipal Lighting Station," by F. W. Ballard; "Low-Pressure Turbines," by a representative of the General Electric Co., and "Attainment and Maintenance of Boiler-Room Efficiency," by a representative of the Harrison Safety Boiler Works.

**The American Boiler Manufacturers' Association** and the National Tubular Boiler Association at a joint meeting held in Pittsburgh on Mar. 29 unanimously approved the Code of Boiler Specifications prepared by the committee appointed for that purpose by the American Society of Mechanical Engineers, and steps were taken toward securing its adoption by the various states. Several of the members and the representative of a prominent boiler-insurance company declared that they should adopt it as their standard, whether compelled to by legislation or not.

**The National Gas Engine Association** is to hold its annual meeting on June 23 and 24 at the La Salle Hotel, Chicago. Reports will be presented by the Standardization, Insurance, Cost Accounting, Legislative and Publicity committees. The following papers are scheduled: "The Data Work," by Prof. P. S. Rose; "Educating the Buyer to Your Type of Engine," by H. G. Diefendorf; "Possibilities of the Farm Lighting Plant," by C. H. Roth; "What of the Kerosene Engine?" by C. E. Bement; and papers by unannounced speakers on "The Future Work of the Association," "How Dealers May Be Induced to Buy for Cash" and "The Magneto of the Future." An accessory exhibition will be arranged, space in which can be secured by addressing the secretary, H. R. Brate, Lakemont, N. Y.

## NEW PUBLICATIONS

COAL GAS RESIDUALS. By Frederick H. Wagner. Published by the McGraw-Hill Book Co., 239 West 39th St., New York, 1915. Cloth; 179 pages; 6x9 in. Price, $2.

Convinced that, even under normal conditions, the recovery of coal-gas residuals is an important means of conserving our natural resources, Mr. Wagner has described German theory and practice in recovering coal gas byproducts and has given estimates of initial and operating costs and of the possible income from the production of tar, naphthalene, cyanogen, ammonia and benzol. Coke is not considered, but is to be treated in a separate volume. Most of the methods outlined were devised by the German chemist Feld, and relate to installations in Germany. Unfortunately, no photographs are given, but the book is illustrated by drawings of the different apparatus required.

## BOOKS RECEIVED

PRACTICAL IRRIGATION AND PUMPING. By Burton P. Fleming. John Wiley & Sons, Inc., New York. Cloth; 226 pages, 5½x8¼ in.; 27 illustrations; tables. Price, $2.

HEATING AND VENTILATING BUILDINGS. By R. C. Carpenter. John Wiley & Sons, Inc., New York. Cloth; sixth edition; 598 pages, 6x9¼ in.; 290 illustrations; tables. Price, $2.

HEAT ENGINEERING. By Arthur M. Greene, Jr. McGraw-Hill Book Co., Inc., New York. Cloth; 462 pages, 6x9½ in.; 198 illustrations. Price, $4.

CENTRIFUGAL PUMPS. By R. L. Daugherty. McGraw-Hill Book Co., Inc., New York. Cloth; 192 pages; 6x9½ in.; 111 illustrations; tables. Price, $2.

VOCATIONAL MATHEMATICS. By William H. Dooley. D. C. Heath & Co., New York. Cloth; 341 pages; 5x7½ in.; illustrated; tables. Price, $1.

## TRADE CATALOGS

McNab & Harlin Mfg. Co., 55 John St., New York. Bulletin. Brass fittings. Illustrated, 5x7 in.

Trill Indicator Co., Corry, Penn. Booklet. Outside spring engine indicator. Illustrated, 16 pp., 6x9 in.

The Scranton Pump Co., Scranton, Penn. Bulletin No. 101. Duplex piston pumps. Illustrated, 16 pp., 6x9 in.

Armstrong Cork Co., Pittsburgh, Penn. Folder. Nonpareil high-pressure covering for boilers, etc. Illustrated.

"Gripwell" Pulley Covering Co., Candler Building, New York. Folder. Gripwell pulley covering. Illustrated.

The Lagonda Mfg. Co., Springfield, Ohio. Catalog W-1. Lagonda locomotive arch tube cleaners. Illustrated, 12 pp., 6x9 in.

York Mfg. Co., York, Penn. Booklet. Ice-making and refrigerating machinery, ammonia fittings and supplies. Illustrated, 20 pp., 6x9 in.

Automatic Steam Trap & Specialty Co., Detroit, Mich. Catalog No. 8. Barton expansion automatic steam trap. Illustrated, 16 pp., 3½x6 in.

Hercules Float Works, 200-10 Franklin St., Springfield, Mass. Catalog. Seamless copper floats and air chambers. Illustrated, 8 pp., 4x8½ in.

Harbison-Walker Refractories Co., Pittsburgh, Penn. Catalog. Silica, magnesia, chrome and fire clay brick, etc. Illustrated, 160 pp., 4x6½ in.

Yarnall-Waring Co., Chestnut Hill, Philadelphia, Penn. Bulletin R. A. Richards unloaders for air compressors. Illustrated, 8 pp., 6x9 in.

Chicago Pneumatic Tool Co., Fisher Building, Chicago, Ill. Bulletin No. 34-M. Class O steam and power driven compressors. Illustrated, 36 pp., 6x9 in.

## BUSINESS ITEMS

Swift & Co., Chicago, Ill., have recently placed an order with the Builders' Iron Foundry, Providence, R. I., for an extra heavy 4-in. Venturi meter tube with type M register-indicator-recorder for use on their boiler feed service.

The Ingersoll-Rand Co., 11 Broadway, New York, has just issued Form 3015, "Portable Air Compressors," which is a 32-page illustrated treatise on the subject of portable air-compressing outfits. Copies are mailed free on request.

The American District Steam Co., of North Tonawanda, N. Y., has opened offices at Suite 610, West Street Building, 140 Cedar St., N. Y. This office will be in charge of G. C. St. John, formerly president of the New York Steam Company.

The Elliott Co., 6908 Susquehanna St., Pittsburgh, Penn., has published two new leaflets of interest to power-plant men, one on twin oil strainers, type "U," and the other (Bulletin J) on pump governors. Copies will be sent on application to the company.

J. G. De Remer, member of the American Society of Mechanical Engineers, American Institute of Electrical Engineers, and formerly chief mechanical and electrical engineer of the United Light & Power Co., San Francisco, has been made manager of the general engineering department of the American District Steam Co., of North Tonawanda, N. Y.

The Canadian Fairbanks-Morse Co., Ltd., has been appointed selling agent in Canada for Penflex metal hose, manufactured by the Pennsylvania Flexible Metallic Tubing Co., Broad and Arch St., Philadelphia. The Canadian Fairbanks-Morse Co. has branches in 14 of the largest cities of Canada, and will carry a complete stock of Penflex in different sizes and styles at its different warehouses, so that Canadian consumers will have the same service as purchasers on this side of the boundary.